ENERGY CONSERVATION IN PRESENT AND FUTURE SHIPS

TP 6855E

For

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17. Abstract

This reference document provides a compendium of information on current practice and potential future developments in the area of energy conservation on ships. It presents information on fuel quality and equipment, propulsion power systems, thrust system, auxiliary systems, ship resistance, ship operation, the application of fuel efficiency measures to various Canadian ships, future fuels and power sources, and future propulsion systems and equipment. It discusses existing technology and future developments with special reference to the application of this technology to Canadian vessels. The text is complimented by the use of graphics. Previously published information is fully referenced and a complete list of references is provided.

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17. Résumé

Cet ouvrage de référence constitue un compendium sur les techniques actuelles et à l'étude en matière d'économie d'énergie à bord des navires. L'information fournie concerne: la qualité des combustibles et les systèmes d'alimentation, les installations motrices, le système propulsif, les systèmes auxiliaires, la résistance à l'avancement du navire, l'exploitation du navire, les mesures d'économie d'énergie appliquées à bord de divers navires canadiens, les combustibles et les installations motrices de l'avenir ainsi que l'évolution future des systèmes et des équipements de propulsion. De plus, il traite des technologies actuelles et futures, en général, et de leurs utilisations sur les navires canadiens, en particulier. En plus de la documentation graphique, l'ouvrage est complété d'un index des articles parus et d'une bibliographie complète.

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E.W. Thompson
Project Manager

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1.0 INTRODUCTION

In the years prior to 1973, the cost of fuel amounted to less than 8% of a ship's operating costs. Since that time the price of fuel oil has progressively risen to 7 times the pre-1973 figure and fuel costs now amount to more than 30% of the ship's operating costs. Figure 1.1 depicts in graphical form the increasing importance of fuel costs.

In addition, many analysts have predicted that natural oil will start to be depleted by the year 2000. This combination of circumstances has encouraged the worldwide trend towards improving the existing efficiency of ships.

This report documents a study carried out by Melville Shipping Ltd. (Melville) as a cost shared project with the Transportation Development Centre (TDC). The study's principal objectives were to review the development of marine technology for the future energy-efficient ship and to forecast the most probable future developments. The study is intended to be of particular interest to shipowners and shipbuilders by providing a guide to the choice of marine power plant, and to those in Government responsible for stimulating marine research and development.

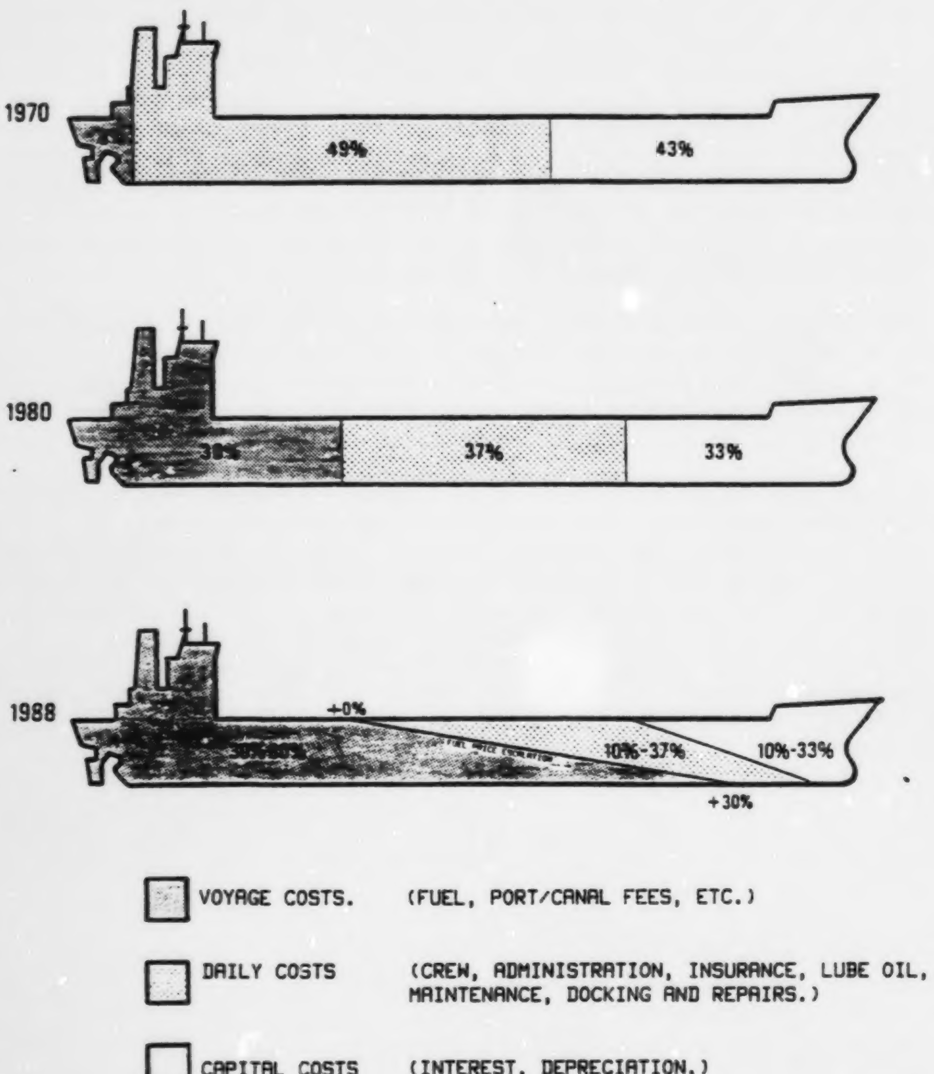
In the course of the study, various important manufacturers of marine equipment outside Canada have been visited and shipowners in Canada and overseas have been consulted. Considerable use has been made of modern literature which discusses energy efficiency and alternative fuel technology and extensive references are to be found in this report.

The report has been organized in a logical sequence commencing with the discussion of modern oil fuel systems and methods of conditioning fuel oils for use in present day engines followed by a review of present day propulsion, thrust and auxiliary systems. The subject of ship resistance and ship operation has been addressed as well as the application of modern systems to various ship types commonly found in the Canadian fleet. Areas for future research and development have been suggested and included in this section of the report, alternative fuels suitable for shipboard use have been identified together with some of the equipment necessary to utilize those fuels.

FIGURE 1.1

GROWTH OF FUEL COSTS IN TERMS OF TOTAL SHIP OPERATING COSTS

(TYPICAL 170,000 DWT BULK CARRIER)



[1]

2.0 FUEL QUALITY AND EQUIPMENT

2.1 General

Since the 1950's oil engines have been developed which can efficiently burn the lowest grade of bunkers available, i.e., residual fuels. However, it is generally predicted [2] that the quality of marine bunkers will deteriorate in the coming years and the ability of oil engines to burn the residual fuels of the future is unknown.

This section explores this area of fuel quality and its impact on equipment requirements.

2.2 Fuel Classification

The general classification for marine fuel is at present based upon viscosity, and the price of marine fuel changes with this parameter as shown on Figure 2.1.

Since only a small part of the worldwide fuel production is used in marine diesel engines;

- 4% of total worldwide petroleum products, or
- 15% of total worldwide production of residual fuel

there is little incentive for the refiners to take into consideration the fuel quality requirements of marine engine builders. Problems of fuel quality can be partly solved by adoption of an international minimum standard for marine bunker oils by major oil companies and bunker suppliers. Such standards have already been proposed. The world's first standard (BS MA 100) shown on Table 2.1 [3] has been published by the British Standard Institution (BSI). This standard is similar to the specification issued by the International Council of Combustion Engines (CIMAC) and to the draft specification currently being evaluated by the International Organization for Standardization (ISO). Unfortunately at present the bunker oil supply companies may not undertake to supply oil fuel to the standards.

A brief list of fuel characteristics and their effects on the engine as well as remedial measures which may be taken are shown by Table 2.2.

2.3 Fuel Cleanliness

Fuel cleanliness is a major factor affecting efficent combustion. The fuel oils of the future, due to their poor quality, must be adequately treated and supplied to the engine in a condition which will ensure high reliability and minimum maintenance requirements of the engine. Some treatment methods are already widely used, and others are at the development stage [4]. Treatment methods which are commonly used include:

- centrifuging
- filtration
- homogenisation

FIGURE 2.1

FUEL OIL PRICE AS A FUNCTION OF VISCOSITY
(MARCH 1983 VALUES)

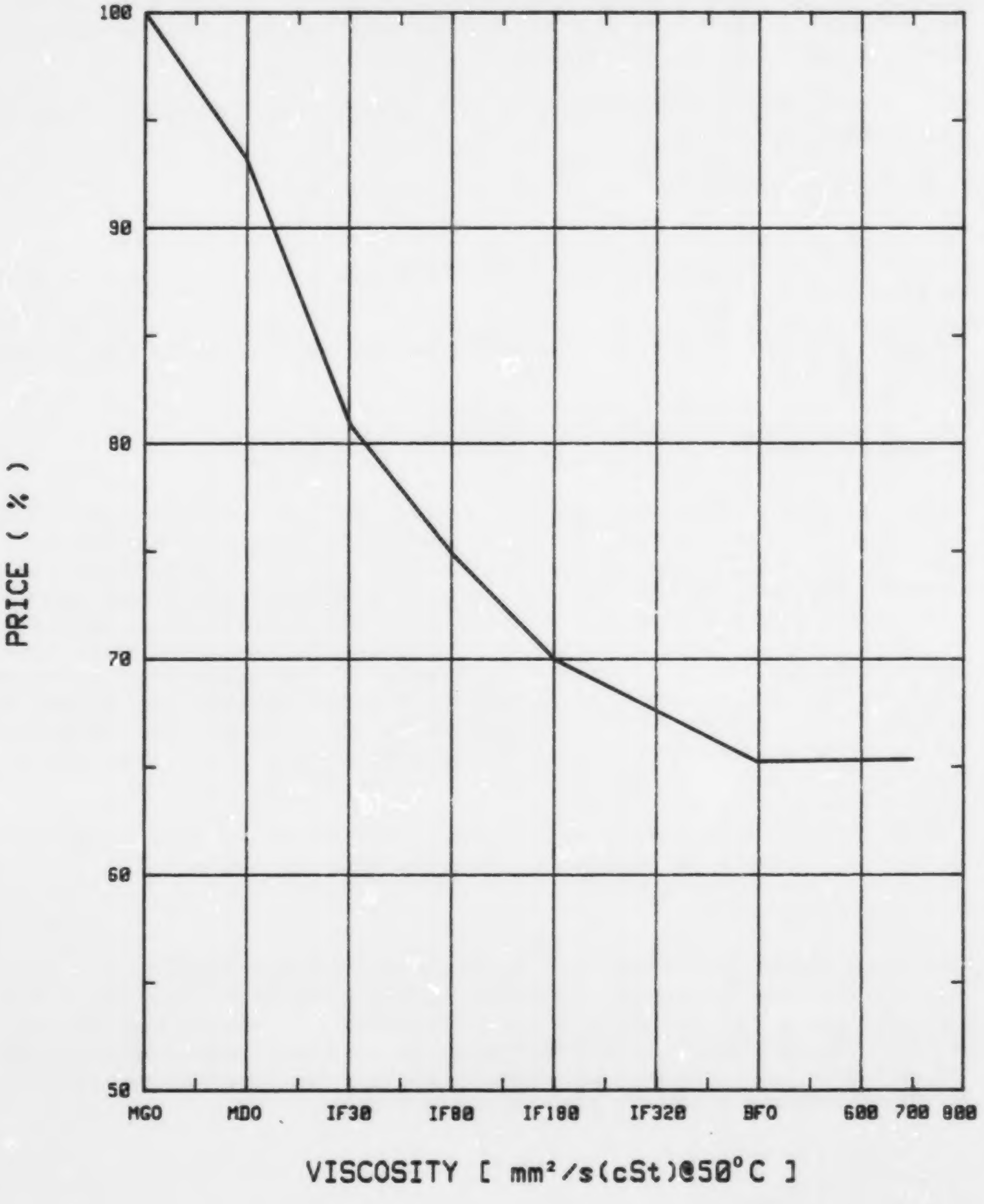


TABLE 2.1

BRITISH STANDARD INSTITUTION - B.S. M.A. 100

Property	Class M1	Class M2	Class M3	Class M4	Class M5	Class M6	Class M7	Class M8	Class M9	Class M10	Class M11	Class M12
Density at 15°C, g/mL, max	—	0·9000	0·9200	0·9910	0·9910	0·9910	0·9910	0·9910	0·9910	—	—	—
Viscosity, kinematic, at 40°C, cSt.*												
min	1·50	—	—	—	—	—	—	—	—	—	—	—
max	5·50	11·00	14·00	—	—	—	—	—	—	—	—	—
Viscosity, kinematic, at 80°C, cSt.* max	—	—	—	15·00	25·00	45·00	75·0	100·0	130·0	75·0	100·0	130·0
Cetane index, min	45	35	—	—	—	—	—	—	—	—	—	—
Carbon residue, Ramsbottom, % (m/m), max	—	0·25	2·5	—	—	—	—	—	—	—	—	—
Carbon residue, Ramsbottom on 10% residue, % (m/m), max	0·20	—	—	—	—	—	—	—	—	—	—	—
Carbon residue, Conradson, % (m/m), max	—	—	—	12·0	14·0	20·0	22·0	22·0	22·0	—	—	—
Flash point, closed, Pensky-Martens, °C, min	43·0	60·0	60·0	60·0	60·0	60·0	60·0	60·0	60·0	60·0	60·0	60·0
Water content, % (V/V), max	0·05	0·25	0·30	0·50	0·80	1·0	1·0	1·0	1·0	1·0	1·0	1·0
Sediment by extraction, % (m/m), max	0·01	0·02	—	—	—	—	—	—	—	—	—	—
Ash, % (m/m), max	0·01	0·01	0·05	0·10	0·10	0·15	0·20	0·20	0·20	0·20	0·20	0·20
Sulphur content, % (m/m), max	1·00	2·00	2·00	3·50	4·00	5·00	5·00	5·00	5·00	5·00	5·00	5·00
Cloud point, °C, max	−16	—	—	—	—	—	—	—	—	—	—	—
Pour point, upper†, °C, max												
(1 December to 31 March)	—	0	0	24	30	30	30	30	30	30	30	30
(1 April to 30 November)	—	6	6	24	30	30	30	30	30	30	30	30
Vanadium content, mg/kg, as V, max	—	—	100	250	350	500	600	600	600	600	600	600

*1 cSt = 1 mm^2/s. †The word 'upper' does not apply to classes M2 and M3.

Other properties for which requirements will be specified when test methods are developed

Property	Class M1	Class M2	Class M3	Class M4	Class M5	Class M6	Class M7	Class M8	Class M9	Class M10	Class M11	Class M12
Ignition quality, min	—	—	*	*	*	*	*	*	*	—	—	—
Total sediment, max	—	—	†	†	†	†	†	†	†	†	†	†
Aluminium content, mg/kg, max	—	—	30	30	30	30	30	30	30	—	—	—

*A limit for this property is considered to be important, but no suitable method of test is available.
†A value is to be agreed on the basis of a method which is in course of development.

[3]

TABLE 2.2

FUEL CHARACTERISTICS

CHARACTERISTICS		EFFECT ON THE ENGINE	REMEDIAL MEASURES
Density (15°C)	g/ml	As the density increases, the separating effect decreases. More corrosive and abrasive constituents (inter alia, water, sand, rust, catalyst residue) enter the engine. High wear, thick ash deposits.	Preliminary cleaning in settling tank, higher separating temperature, lower flow rate, recirculation, increasing weight of sealing water by means of additives.
Water	Vol. %	Corrosion and steam bubble formation in system, atomization, hot corrosion with sea water.	See density - remedial measures. Slightly raised delivery pressure.
Ash, rust, sand, Al_2O_3 SiO_2	Weight % mg/l	Abrasive wear in fuel system and engine, especially if catalyst residue in fuel.	preliminary cleaning in settling tank, lower rate of flow though separator, wear-resistant material.
Vanadium Sodium	mg/l mg/l	Deposit formation esreclily on exhaust valve seat and in turbo-charger, hot corrosion (largely depending on vanadium:sodium ratio).	Cooled valve seat, rotocap, rotary vane valve, water-washing fuel, fuel additive, separation so that the water is largely separated out.
Viscosity	cSt/50°C	Preheating the fuel to less than 150°C is conducive to asphaltenes formation; effects see asphaltenes. An excessive injection viscosity results in overloading of pump drive gear and injection pipes as well as poor atomization. Too low a viscosity also results in poor atomization.	Increasing capacity of final preheater so that specified injection viscosity can be reached. Ordering only fuel with a viscosity high enough for preheating to less than 150°C. Viscosimeter which automatically sets viscosity specified.
Carbon residue (Conradson)	Weight %	Formation of carbon deposits by incomplete combustion, expecially at part load. Abrasive wear, reduced heat transfer to cooling water, clogging of rings.	Good mixture formation, atomization, efficient component cooling. Further remedial measures see asphaltenes.
Asphaltenes	Weight %	Slow heat release, in extreme cases close to the wall, increased deposit formation.	Electronic injection, air and cooling water temperature control as a function of load. Fuel additives, lubricating oil with adequate cleaning and dispersing capacity.
Incompatibility	Vol. %	C/H balance of fuel mixture is upset. Results are asphaltenes precipitation which, in turn, results in sludge formation in the fuel system, interference with operation of injection system anad thus poor atomization as well as combustion with a high residue level.	Reputable oil companies only use mixture components with a balanced C/H ratio. Fuel from one bunkering station only is filled into storage tank and is used up before new fuel is filled in. Fuel oil is only mixed with distillates aboard if compatibility is ensured and a mixer is available, e.g. knife-type mixer, homogeniser. Fuel additive in storage tank.
Sulphur	Weight %	Chemical wear.	Component temperature above dew point of acid (approx. 180°C) at all loads. Good oil distribution, adequate neutralisation capacity of the lubricating oil at right time.
Cetane Number		If below 35, ignition delay, steep firing pressure rise, high combustion temperature and increased nitric oxide content of exhaust gas are to be expected.	Fuel injection matched to ignition properties (electronic injection), air and cooling water preheating in part load operation. Fuel additives.

[Ref. MAN Technical Instructions]

- blending
- emulsifying
- chemical treatment

The engine builders consider that centrifuging is the most suitable method for treating present day fuels prior to delivery to the engine. The other forms of treatment tend to be regarded as supplementary to centrifuging rather than a replacement.

2.3.1 Centrifuging

The solid contaminants in marine fuel oil are mainly rust, sand, dust and refinery catalytic fines (AL_2O_3 and SiO_2). Liquid contaminants are mainly fresh or salt water and hydrocarbon sludge. Effective cleaning can only be ensured by means of a centrifuge. To obtain optimum cleaning it is of the utmost importance to operate the centrifuge with the fuel oil viscosity as low as possible and allow the oil to remain in the centrifuge bowl as long as possible.

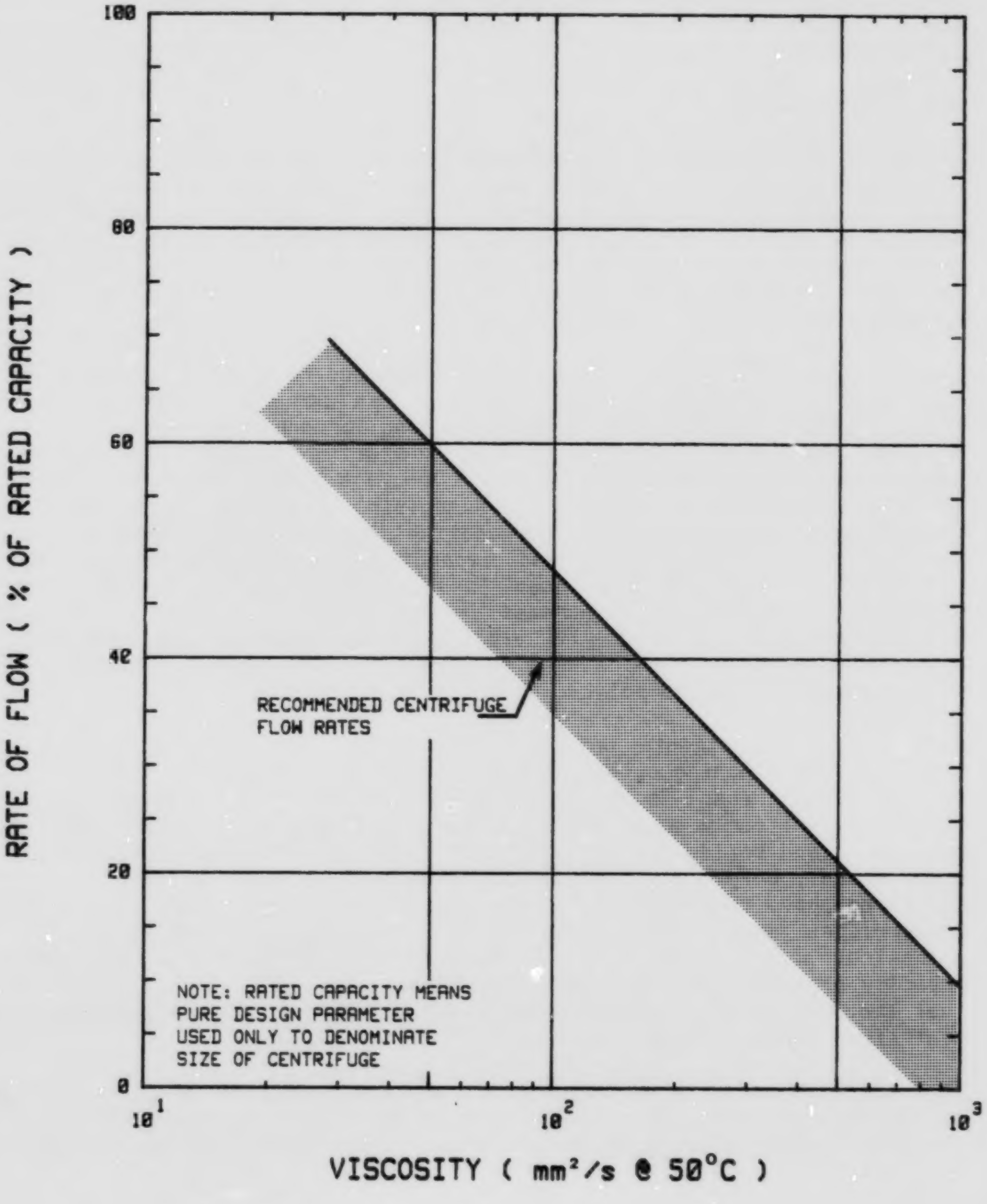
Fuel oil viscosity is controlled by heating, and to avoid evaporation of the seal water in the centrifuge, the oil can not be heated above 98°C. Therefore oils of about 380 cSt/50°C passing through traditional centrifuges are at about the limit that can be heated to attain the recommended separating viscosities below 50 cSt. Oil with viscosity at the centrifuge above 50 cSt can be treated by decreasing the flow rate through the centrifuge although this will increase the size of separating equipment [5]. The recommended flow rate and separation temperature plotted against viscosity of the fuel are shown in Figures 2.2 and 2.3. As fuel oil viscosity is expected to increase in the future the cost of centrifuge equipment for future ships will also increase, by about 70%, due to lower oil flow rates which must be used to attain the same degree of separation of the unwanted fluids and solids.

Results from experimental work on treating RFO by centrifuging have shown that the best cleaning effect, especially for the removal of catalyst fines, is achieved when separators are operated in series, in the purifier-clarifier mode as shown in Figure 2.4 [6, 7]. The purifier removes both water and solids from the oil and the clarifier, which operates without a water seal, only the solids. This method allows the clarifier oil feed to be heated above the temperature permissible for the purifier. Clarification temperatures are limited only by the temperature restriction of the mechanical seal, gearbox oil or cracking of the fuel oil. The operating capacity of each individual centrifuge should handle the total amount of the fuel required by the engine, without exceeding the flow recommended by the centrifuge maker for the operating mode which is used.

The installation of centrifuges which are too small, in order to economise on the capital cost, is probably the reason that some manufacturers claim the best system is parallel purification. This should not be considered as good practice.

FIGURE 2.2

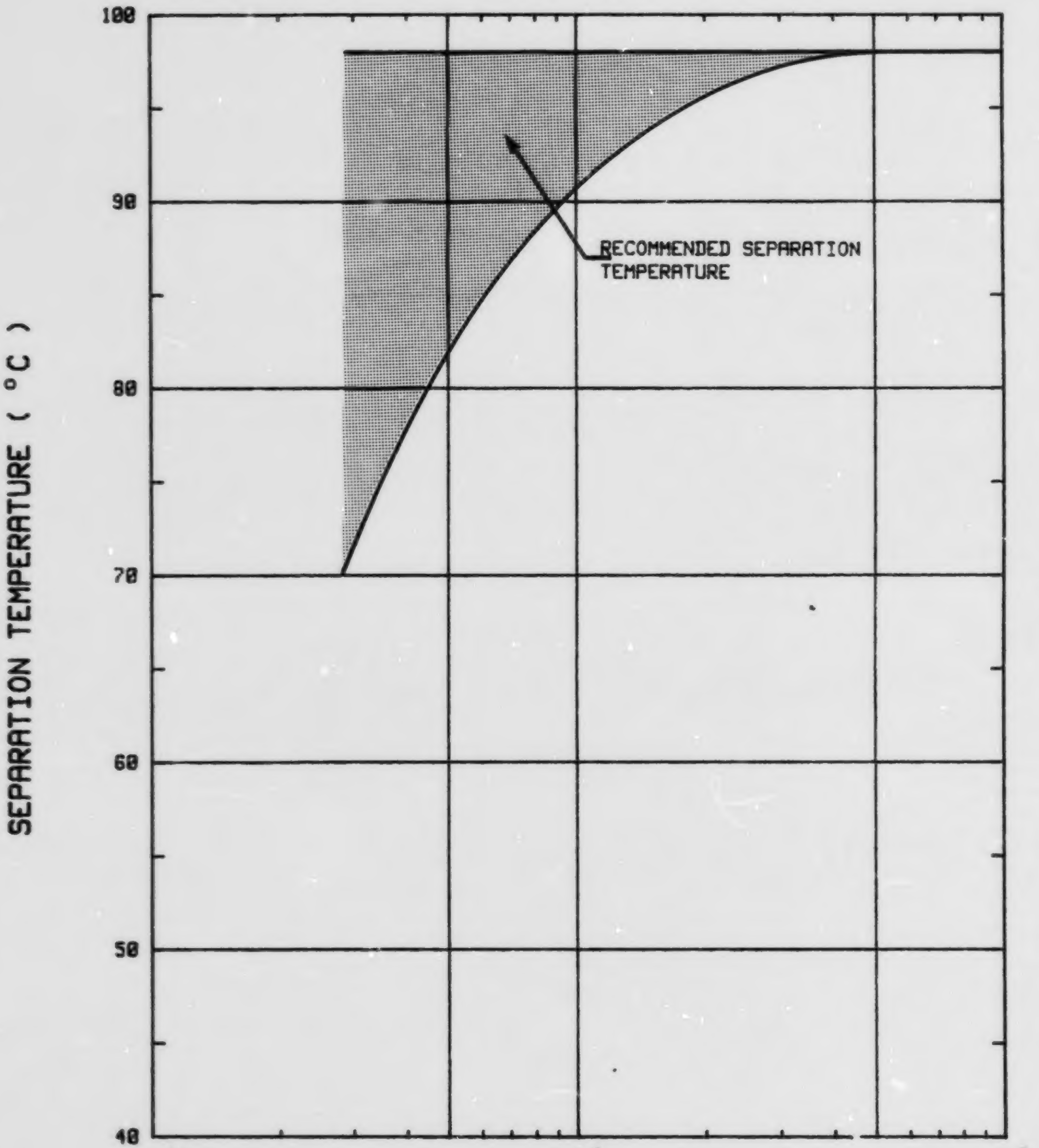
CENTRIFUGE FLOW RATE VS. OIL VISCOSITY



[5]

FIGURE 2.3

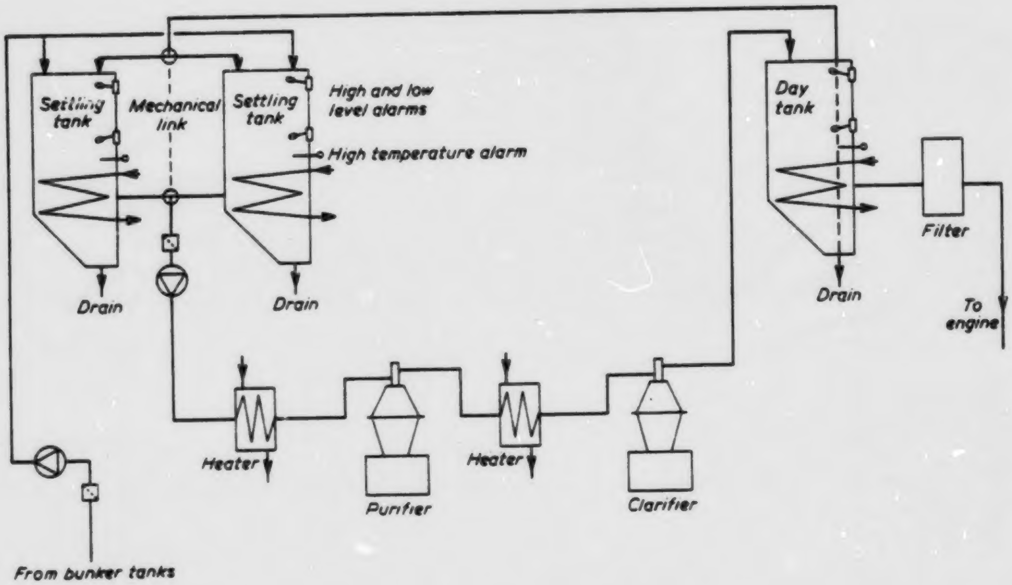
CENTRIFUGE SEPARATION TEMPERATURE VS. OIL VISCOSITY



[5]

FIGURE 2.4

SERIES METHOD OF FUEL PURIFYING


Settling tank
Mechanical link
Settling tank
High and low level alarms
High temperature alarm
Day tank
Filter
Drain
Drain
Drain
To engine
Heater
Purifier
Heater
Clarifier
From bunker tanks

[4]

The most significant problem associated with deteriorating fuel quality, is bunkers with densities above 0.99 g/cm^3. Bunkers with densities approaching that of water cannot be separated from the water by the normal centrifuging process and a number of solutions have been investigated, in particular those based on the use of products for rendering the water seal more dense. The Alfa-Laval Co. has carried out tests using solution of magnesium sulphate, with a density 1.2 g/cm^3 (see Figure 2.5) which can be adopted if adequate monitoring and control are developed for the equipment and if the process can be made suitable for use by ship's staff [8]. Initial experiments have suggested that while this system is technically feasible, there are practical operating problems to be overcome before it is suitable for general use.

Some engine manufacturers permit water to be retained in the fuel, provided that the proportion of sea water content is low due to the damaging affect of sodium. An excess of water in the fuel requires the use of a homogenizer which breaks the water down to droplet size of about 5 microns.

To ensure good purification it is vital to maintain a steady fuel flow rate and a stable temperature at the purifier. In this respect the following should be given particular attention.

- Pumps and pipes should be designed to reduce emulsification.
- Thermostats should be fast acting; this is most important during desludging.

Increased sedimentation means greater sludge rejection by the purifier/clarifier. The resultant loss of combustible hydrocarbons represents a significant loss of energy and should be avoided. The energy contained in the sludge, although not suitable for use in an internal combustion engine, can be recovered in a boiler.

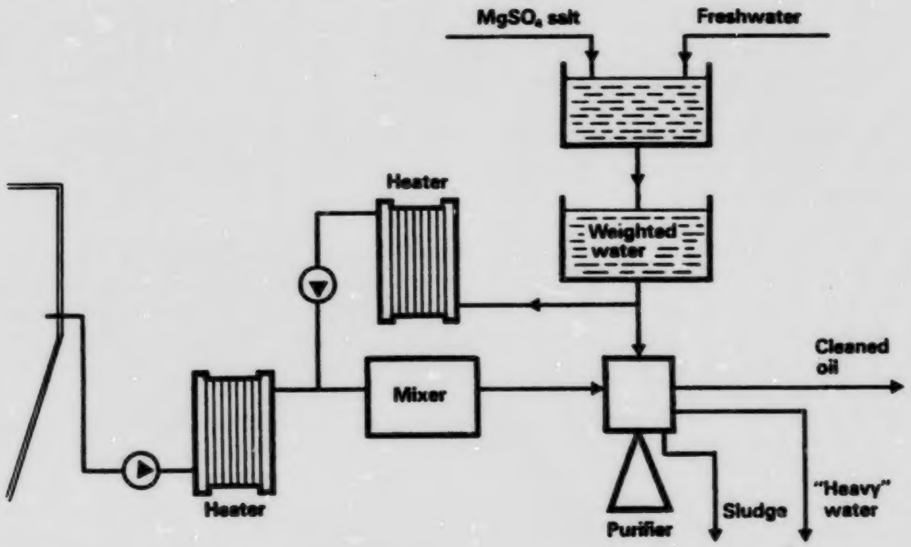
The latest models of centrifuges for cleaning heavy fuel oil, contain a water transducer which monitors changes in the water content in the cleaned oil. The increase of water content of the fuel indicates that there is too much water in the bowl and a signal from the water transducer to a microprocessor initiates discharge [9].

Fuel loss from a purifier due to loss of bowl seal can be significant. Existing methods of containing these losses are not effective enough but can be improved if required. At present shipowners do not appear to consider the extra investment in more efficient "catchment" systems justified.

The rejected sludge from the engine fuel oil purifier system can be used as boiler fuel even in existing boiler installations. Several vessels have been supplied with sludge treatment units which heat the sludge, separate water and blend the residue with heavy fuel in a high-shear mixer. The value of the heat recovered in the boiler is said to be equivalent to about 0.7% of the fuel passing through the purifiers. A rotary cup burner system is best suited for this purpose [10, 13].

FIGURE 2.5

WEIGHTED METHOD OF HEAVY FUEL PURIFYING



MgSO$_4$ salt
Freshwater
Heater
Weighted water
Cleaned oil
Mixer
Heater
Purifier
Sludge
"Heavy" water

[8]

2.3.2 Filtration

Filtration is not regarded by most marine diesel engine manufacturers as a satisfactory fuel treatment system, especially for heavy fuel burning engines. It is probably not practicable to install filters of the required capacity to remove the relatively large amounts of contaminants and water found in heavy fuels. Due to the high fuel consumption of large modern engines, filter bodies would be very large, and hence expensive, and cartridge life would shorten to unmanageable periods. However, modern filters have some advantages and in the future may be developed to compete successfully with centrifuges. The choice of suitable filters is restricted.

A self-cleaning filter or strainer is not of great value because it can retain only particles larger than the mesh and if fine meshes are used, it becomes increasingly difficult to achieve successful self-cleaning operation as the filter chokes too rapidly. The pleated depth cartridge filter overcomes some of these disadvantages as it is capable of retaining a wide spectrum of particle sizes. It has proved to be largely unaffected by the presence of asphaltenes in the fuel and it is simple to service. Figure 2.6 shows filter performance compared with centrifuge trial results and demonstrates the main advantage of filters which is the collecting of substantial quantities of catalytic fines.

Most important is the quality of the filter seals and the strength of the filter material since the differential pressure across the filter can reach 10 bar. Rigid control must be maintained over the fuel temperature to prevent damage to the filter, opening the way to destructive catalytic fines entering the engine. Engineering prudence would suggest that filters could most effectively be used as safeguards after the primary fuel treatment by centrifuges after the bulk of particulate matter has been removed. The use of filters positioned immediately before the engine is recommended by most diesel engine manufactureres [11].

2.3.3 Homogenisation

The purpose of homogenisers is simply to mix all the constituents very thoroughly, and grind any solids into fine particles.

Residual fuel oil also contains agglomerates, i.e. collected particles of asphaltenes, which are often up to 200 microns in size. Asphaltenes are slow burning and are not completely reduced in the cylinder, but form a sticky mixture that is deposited on the pistons, cylinder liners, fuel pumps and injector nozzles. Homogenisation breaks down the agglomerates and disperses any water content within the fuel leaving the impurities equal in size to the molecules of the base oil. This provides a more uniform substance for the engine to burn [12].

The most popular homogeniser passes the fuel oil through a housing where a series of rolling discs subject the fuel to a shearing and crushing force. However, there is considerable doubt of the ability of homogenisers to render the sediment and dirt, and especially the very abrasive catalytic fines, harmlessly small. Therefore, with fuel having a viscosity over

180 cSt, the homogeniser should be used in conjunction with a purifier. By the use of a smaller gravity disc on the centrifuge the oil will retain some of the heavy constituents that would normally be taken out as sludge. The saving of some of this combustible material is the main advantage of the homogenisation [14].

In Japan the latest homogeniser uses ultrasonic waves to blend the fuel. This unit consists of an ultrasonic horn and a cylindrical rotary filter coated with 5 micron mesh through which fuel flows from the outside to the inside. Some special features claimed for this homogeniser are; a complete elimination of asphaltic sludge, superior filtration of particles greater than 5 microns, more effective sludge disposal where differences in specific gravity of sludge and fuel oil cause difficulties in centrifugal separation. It is said that 100 units of this kind have been fitted to ships since 1979, all of which have given successful results. The largest unit presently available is suitable for use with an engine with a fuel consumption of 50 tonnes/day. This ultrasonic fuel oil homogeniser, called "Marisave", is produced by the Sanko Marine Co. It is presumed that much will be learned from the systems now in use as a means of assessing their future value [13].

2.3.4 Blending

Fuel blending is principally for use with auxiliary engines to permit them to operate on a heavier and less expensive grade of fuel. A fuel blender allows the operator to blend fuel on board and produce oil of desired viscosity from a heavier and lighter grade. Blending is now being gradually accepted as a way of reducing total costs. Today, well over 2,000 ships have been equipped with onboard blenders and it is claimed many are recording savings of $50,000 to $60,000 per year [15, 16]. These savings far outweigh the increase of maintenance costs of the blender and those associated with the burning of heavy fuel oil. Figure 2.7 shows the prices of blended fuel compared to the percentage of diesel oil in the blend.

The quality and parameters of blended oil acceptable to an engine vary widely according to engine type, manufacturer's recommendations, running conditions and engine speed. Typical blends in use for auxiliary engines and main engines during manoeuvring are [16]:

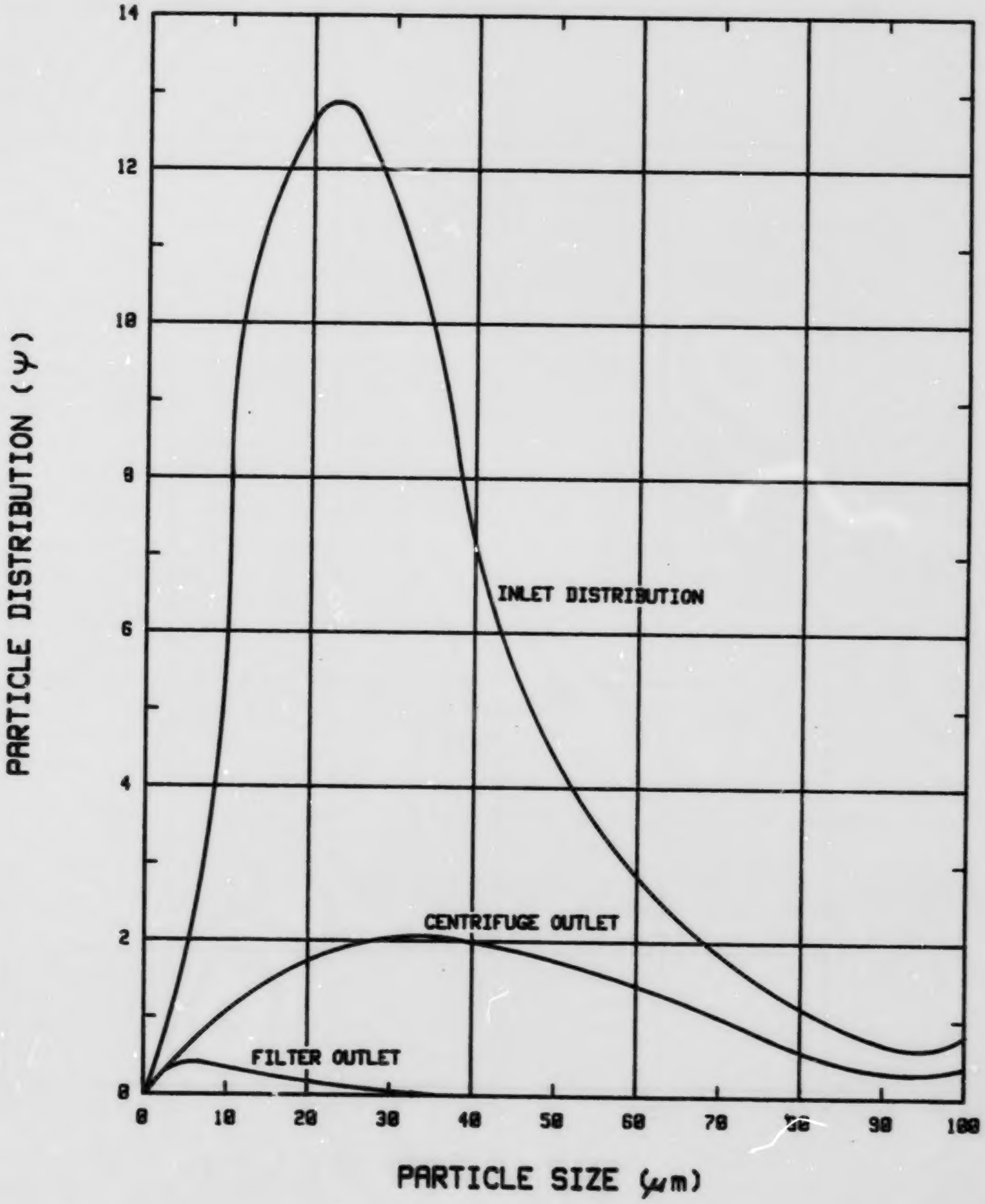
for HO 110 - 130 cSt	MDO : HO = 50 : 50
for HO 180 - 240 cSt	MDO : HO = 60 : 40
for HO 270 - 320 cSt	MDO : HO = 70 : 30

The blending ratio necesssary to obtain the desired viscosity of blended fuel can be determined by means of a blending chart, as shown in Figure 2.8.

The mixing of fuels is accomplished either "dynamically" by mechanical vane agitation, which is mostly used in "batch oil" systems or "statically" by hydrodynamically forcing the two oil flows into turbulence in a baffled chamber which is used in "in-line" systems. Dynamic blenders seem to be preferable to static devices but the latest innovation of incoporating into the blending unit a very effective homogeniser also gives good performance to static blenders. Many older types of blenders do not homogenise the two fuels but merely mix them, causing problems for incompatible fuels.

FIGURE 2.6

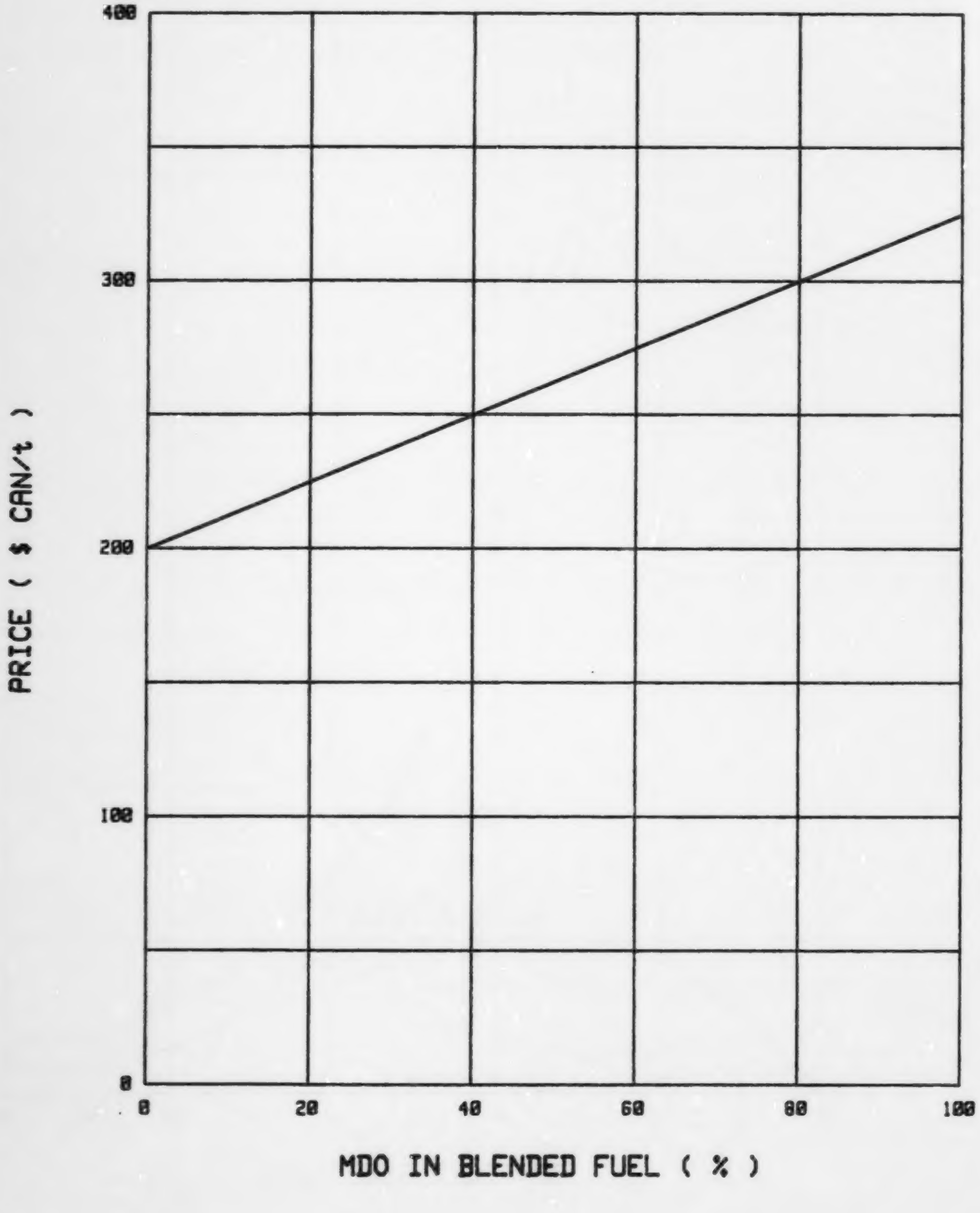
COMPARISON OF FILTER AND CENTRIFUGE PERFORMANCE



[11]

FIGURE 2.7

PRICE OF BLENDED FUEL
VS. PERCENTAGE OF DIESEL OIL CONTENT

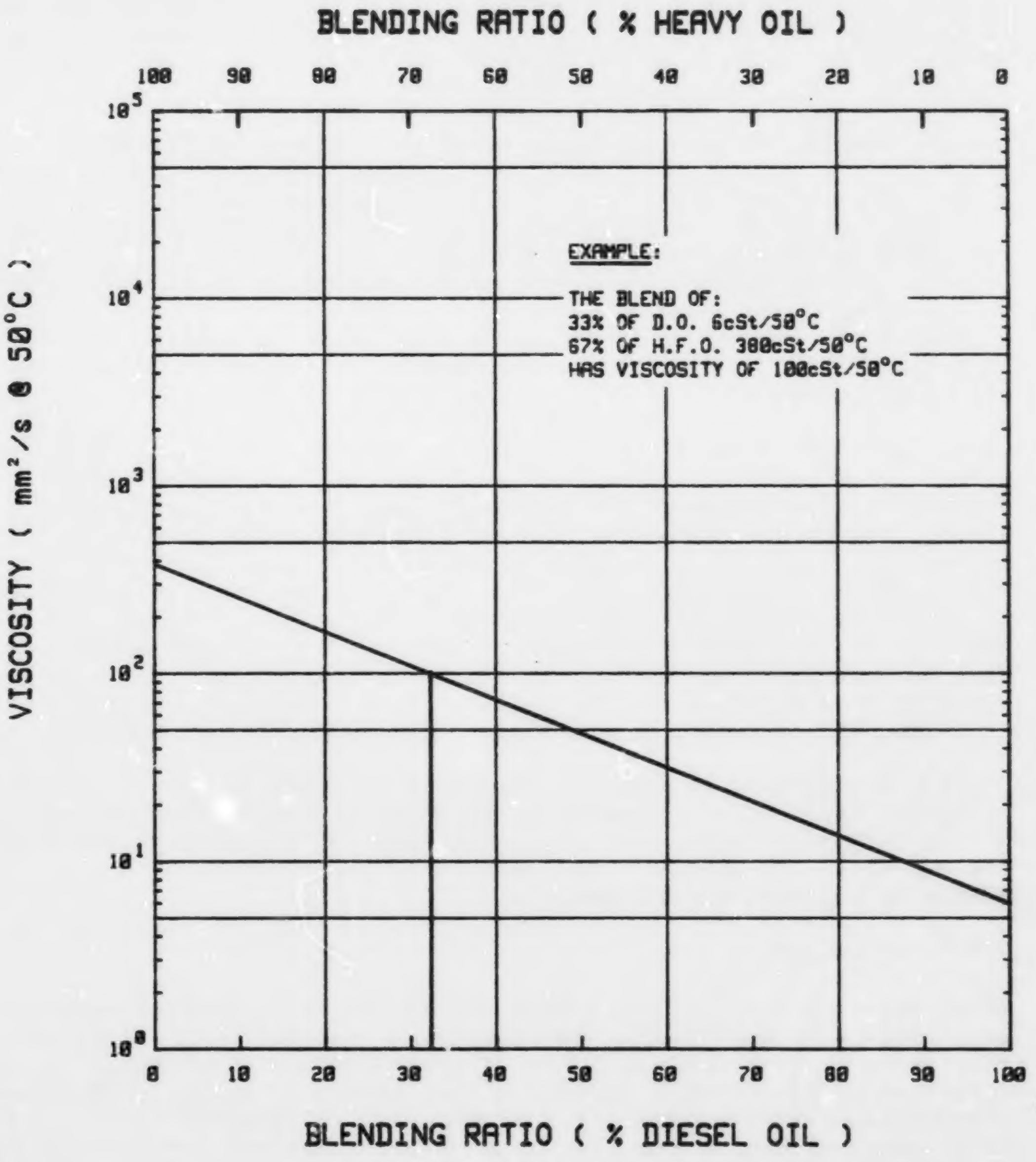


DATA AT MARCH 1983

MELVILLE SHIPPING LTD

FIGURE 2.8

FUEL BLENDING CHART



EXAMPLE:

THE BLEND OF:
33% OF D.O. 6cSt/50°C
67% OF H.F.O. 380cSt/50°C
HAS VISCOSITY OF 100cSt/50°C

[16]

Effective homogenisation to reduce the asphaltene particles to 5 microns diameter, makes the fuel a stable emulsion. Dynamic homogenisers will also cater for small quantities of water and overcome some of the problems normally associated with water contamination [17].

A special blending system called "Bunkermaster" has been designed by Dodwell Industrial, Tokyo for coastal carriers and small ships with medium speed main propulsion engines. Such ships normally do not have heating coils in the bunker tanks which prevents the bunkering of heavy fuel oil and blending on board. The "Bunkermaster" system installed in the bunkering line on the deck allows heavy fuel oil to be bunkered directly to the vessel with marine diesel oil (already on board) injected at the proper volume to achieve a blended oil viscosity of around 20-30 cSt/50°C.

In-line blending of fuel oil has a number of claimed advantages:

- A separate tank for blended fuel is not needed.
- Temperature and hence fuel viscosity can be precisely controlled by an automatic viscosity control unit.
- Any desired blend ratio can be instantly obtained by adjusting the setting of the metering pump.
- When the engines are running on light load more diesel oil is introduced into the blend and automatically reduced as the engine load increases.

Some fuels are incompatible and cannot be blended, as large quantities of sludge and sediment will precipitate. Sludge can form at the blender itself and result in blockage of the discharge line. If blending is carried out before treatment, then any sludge generated might result in choked centrifuges. Even if the sludge is manageable at the centrifuge, problems can arise in the service tank. To avoid problems at the centrifuge or service tank, it is usual to blend before the engine. In that case sludge can form at the engine filters and if some asphaltic sludge passes with the fuel to the injection system, severe combustion problems can result with disastrous consequences for the engines. This incompatibility between fuels is often cited by owners as the reason for not fitting fuel blenders.

It is also important to note that contaminants cannot be blended out of fuel and their level is only reduced by dilution. In cases when marine diesel oil percentage of a blend is high, the level of contaminants contained by the residual oil component can fall below the limit specified by engine makers in the resultant mixture.

2.3.5 Emulsification

After some years of trials ashore and at sea with burning water-in-oil emulsions in diesel engines, the results are now published. It is reported that savings of fuel between 3.5 to 4% are achievable under actual operating conditions. Under certain conditions fuel savings in excess of 6% have been claimed when using a water in fuel emulsion of about 8%. The distilled water required can be provided in sufficent quantities from the standard waste heat evaporation unit fitted to the engine cooling water system [4, 18].

The other benefits of fuel emulsifiers are cleaner engines and reduction in toxic nitrous oxide emmission. The turbo charger and exhaust gas side of engines which have operated on oil water emulsions have been reported as considerably cleaner and thus maintenance intervals are extended.

Several ships have already been supplied with emulsifiers. The experience of operation of the equipment on these vessels has showed that there is a slightly larger increase in saving on a slow speed engine using heavier viscosity fuels than on a medium speed engine burning lighter fuels [19].

When water is added to the fuel, the viscosity of the emulsion increases, which is automatically compensated by heating. When a water fuel emulsion is being burnt the engine exhaust gas temperature tends to reduce but any reduction in energy recovered from the exhaust will be offset by increased fuel saving. Fuel injectors and pumps reveal no cavitation, wear or corrosion. Stability of the emulsion is not seen as a problem. A sample of fuel was taken from one ship and analysis 10 days later showed that water particles in the sample were still in the 2 to 4 micron range [20].

To install the emulsifying unit on a vessel with a slow speed engine burning fuel of 360 cSt/50°C, the fuel system must be modified slightly with the addition of an extra booster pump to keep the recirculated oil under pressure in order to prevent water boil off from the emulsion.

The combination of a fuel blender with an emulsifier developed in West Germany and termed Flux-Mix-Dos has been in service for some time on a large bulk carrier and the fuel bill for the auxiliary diesels was reduced by 6.5%. It is said that this system differs from other blenders as it incorporates a high quality homogeniser which breaks the oil to droplets of approximately 2-3 microns diameter. When water is added the emulsion can then be stored and is said to remain stable for up to eight months [21].

There has been open scepticism in the marine industry on the effectiveness of the emulsifier and it is probable that more tests must be carried out before the equipment is fully proven and accepted.

2.3.6 Chemical Treatment

Fuel additives can make a positive contribution to the performance of a vessel's propulsion and power plant. However, the reason for its use must be known, and the proper fuel treatment program implemented and adhered to. Unfortunately, in many instances owners are adding expensive additives to fuel with little real understanding of the processes involved and no objective monitoring of the benefits which are gained. For the benefits of additives to be realised their use must be tailored to the chemistry of the fuel to be burned. This requires onboard testing of each batch of fuel to determine both the type and quantity of the additive to be employed.

By the time International Fuel Standards are commonly used by fuel suppliers, onboard fuel testing will be the most important process to determine the scope and quality of the necessary chemical treatment. The range of available chemicals and their functions is considerable and there are three phases of the fuel burning process that could benefit from the addition of fuel additives:

- Pre-flame, from storage tanks to injection
- Flame, during combustion of the fuel
- Post-flame, after the fuel is burned

Table 2.3 lists the main classes of fuel additives and their functions. However, it is very difficult to establish the benefits and cost effectiveness of using additives. Impartial information is not presently available in sufficient detail on quantity of additives to be used and price, to permit an assessment to be made of the economic benefits to be realised. Engine manufacturers generally dispute the claims made for chemicals used to improve combustion. It is apparent that many owners believe that the use of fuel additives is a substitute for inadequate adjustment and maintenance of their engines [2, 22, 23].

2.4 Fuel Monitoring

Monitoring fuel consumption is a new technology, where many problems have yet to be overcome. The real time measurement of fuel consumption in conjunction with speed and other variables such as rev/min, propeller pitch, distance, etc. will be a powerful tool to enable effective control of energy expended.

There are some available systems under various names such as "econometers", "fuel savers", "efficiency analyzers", etc. All of them are computerized systems which register a variety of vessel conditions and evaluate performance. The instruments display the information and calculate appropriate optimum speed, optimum propeller pitch and fuel consumption.

Transmitters feed information from several input sources to the system's central computer. Usually inputs include:

- fuel flow (with compensation for calorific value)
- speed and distance
- shaft power and r.p.m.
- trim
- real time

Examples of these system are:

- Monitor 205 produced by Avicon Co., Scottsdale, Arizona, U.S.A.
- Kontram Fuel Saver by OY Kontram AB, Helsinki, Finland
- SAL-FE by Jungner Marine, Solna, Sweden
- Fluidyne 1071 and 1072 (for steam vessels) by Fluidyne Instrumentation, Oakland, California, U.S.A.

The measurement of heavy fuel flow is very difficult. Conventional volumetric flow meters, using hydrodynamic principles, are sensitive to variations in fuel properties and positive displacement types usually involve moving parts with tight clearances. All volumetric measuring devices can only obtain the mass rate of flow after taking into account the density of the fuel, a factor that may readily vary by 5% between different fuels. Most recently, a unique weighing system of fuel flow has been developed which avoids the difficulties encountered with conventional flow

TABLE 2.3

DIESEL FUEL ADDITIVES

ADDITIVE CLASS	PROBLEM LOCATION	PRIMARY FUNCTION	SECONDARY EFFECTS
Dispersants	Tanks and Lines	Dissolves sludge allowing free flow of fuel and cleaning of tanks, lines and strainers.	Fuel harder to burn*. Smoke and particulates increased.
Stabilisers	Tanks and Lines	Stops sludge formation and corrosion caused by slow oxidation of fuel.	Fuel harder to burn*. Smoke and particulates increased.
Corrosion Inhibitors	Tank and Lines	Stops corrosion caused by organic acids.	Emissions increased.**
Heavy Metal Scavengers	Tank and Lines	Stops corrosion caused by heavy metal ions in fuel.	Engine deposits and particulates increased.**
Pour/Cloud Point Depressants	Tank and Lines	Prevents solidification of fuel and filter plugging at low temperatures.	**
Emulsifiers	Tank and Lines	Holds trace water in fuel, preventing icing and collection.	**
Emulsion Breakers	Tanks	Drops water to bottom of tank for easy removal.	None.
Biocides	Tank and Filters	Kills slime-forming micro-organisms that live at water-fuel interface.	Engine deposits, particulates, emissions increased.**
Slime Dispersants	Tank and Filters	Disperses micro-organism slime that plugs filters.	**
Atomisers	Engine (injectors)	Lowers surface tension of fuel; better combustion from smaller fuel drops.	Better fuel flow, cleaner exhaust; improved efficiency.
Combustion Catalyst	Engine	Better burning through chemical mixing of air and fuel.	Cleaner exhaust.** Improved efficiency.
Cetane Improvers	Engine	Better burning by addition of easily burned 'high quality' compounds.	Improved efficiency.**
Deposit Cleaner	Engine	Solvents designed to dissolve and burn cylinder and ring deposits.	**
Smoke Suppresant	Exhaust	Reduce smoke; mechanism unknown.	Abrasion due to metal particles in additive.**

* Unless compensated for in the additives's formulation.

** Effect dependent upon specific additive formulation.

meters. This system is unaffected by either the viscosity, temperature of the fuel or motion of the vessel. The new gauge is based upon the measurement of the upward force on a cylindrical float as the fluid rises. Since this depends only on the weight of the displaced fluid, the accuracy does not depend on density. The unit does not provide continuous measurement of fuel flow and is arranged to automatically fill the measurement tank every 10 minutes. The unit has been developed by Plint & Partners Ltd., Wokingham, U.K. [84]. The prototype is at present installed on the Townsend Thoresen ferry FREE ENTERPRISE VIII.

Another important feature of fuel flow measurement is the location of the flow meter. The recommended placement is between the float or mixing tank and service tank as shown in Figure 2.9 [85].

While not strictly "energy conservation", it is important that the shipowner should rigorously check the quality of the fuel "as loaded" and compare the analysis with the declared values on the bunker receipts. In many cases it will be found that the stated specific gravity of the fuel and the water content will differ significantly from the values declared on the bunker receipt which, unless detected, will reflect on the apparent fuel consumption and hence energy efficiency of the ship. It is surprising that relatively little attention is give by shipping companies to the measurement of the quantity and quality of the bunkers received on board, yet this is the basis of all energy efficiency measurements. All bunker measurement systems rely upon ullaging or sounding the tanks. Therefore the daily consumption is completely dependent upon the initial accuracy of the tank calibrations.

2.5 Summary

Over the past several years fuel quality and ship operating procedures have changed drastically. The most important changes observed in fuels available today are:

- higher densities, exeeding 1.0
- more widespread contamination with catalytic fines
- easier precipitation of asphaltanes
- increased ability to emulsity with water

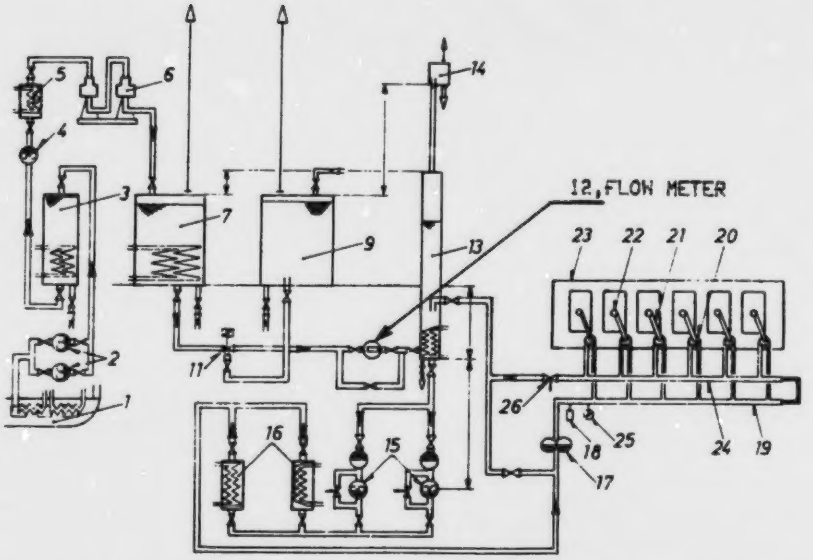
All these properties of fuel oil can have very significant adverse effects on engine wear if not adequately treated. That is why a lot of research work has been carried out to develop modern fuel treatment practices. Since traditionally, heavy fuel oil for use in diesel engines has been treated by centrifugal separators on board ship, the research has gone towards exploration of alternatives to centrifugal separation. Many manufacturers and shipowners experimented with homogenizers and filters, and some even dispensed with the separator altogether owing to the seemingly high capital and maintenance costs.

However, despite numerous benefits claimed in favour of these new methods, recent publications and technical papers have emphasized the importance of the centrifuge, reinforcing the view that all modern diesel vessels burning heavy fuel should be fitted with three separators. The other equipment or treatment should be considered only as supplementary means which help to improve the quality of fuel already cleaned by centrifuging.

Properly designed fuel treatment systems containing recently recommended arrangements, as well as adequate fuel treatment practice, may result in substantial fuel cost savings.

FIGURE 2.9

LOCATION OF FUEL FLOW METER IN ENGINE FUEL SYSTEM


12, FLOW METER
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1	*D B storage tank*
2	*Transfer pump*
3	*Settling tank*
4	*Circulating pump on separator*
5	*Preheater*
6	*Separator*
7	*Heavy oil service tank*
9	*Diesel oil service tank*
11	*Three way valve*
12	*Flow meter*
13	*Mixing tank*
14	*Vapour trap*
15	*Booster pump*
16	*Endheater*
17	*Filter*
18	*Thermometer*
19	*Fuel pump supply pipe*
20	*Fuel injection pump*
21	*High pressure fuel pipe*
22	*Fuel injector*
23	*Engine*
24	*Fuel return pipe*
25	*Pressure gauge*
26	*Constant pressure valve*

[85]

3.0 PROPULSION POWER SYSTEMS

3.1 General

Since the oil crisis of 1973 ships have been designed for operating at lower speed. This is partly due to the change in the relationship of ship operating costs (price of bunkers) and capital cost and partly due to the surplus of tonnage brought about by a decline of world trade. Slower ship speed has resulted in a reduction of the required propulsion power, from the 1973 average of 11 MW per ship to 7.5 MW in 1982 as shown in Figure 3.1.

The number of ships built over the last 12 years is shown in Figure 3.2. More motor ships have been built than those with steam propulsion, in the ratio 10:1 between 1970 and 1976. After 1976 the number of steam ships has fallen to 2 in 1982.

Although wind and gas turbine powered vessels are in operation the types of propulsion machinery almost exclusively used for merchant ships throughout the world are:

- direct drive slow speed oil engines
- geared medium speed oil engines
- geared steam turbines

The characteristics and abilities of oil fired propulsion machinery will be discussed in this section. Other more exotic propulsion systems using gas turbines, COGAS systems, nuclear power, wind power and the utilization of alternative fuel sources will be addressed in other sections.

3.2 Diesel Engines

Modern slow speed engines are now manufactured to develop up to 40,000 kW, or in excess of 3,000 kW per cylinder while medium speed engines can develop in excess of 18,000 kW or more than 1,200 kW per cylinder. Modern slow speed two stoke diesel engines commonly have a specific fuel consumption guaranteed to be less than 180 grams per kilowatt hour and the fuel consumption of medium speed four stroke engines is less than 200 grams per kilowatt hour. Each new type of engine invariably shows a substantially improved SFC over its predecessor. The specific fuel consumptions for slow speed two stroke diesel engines represent thermal efficiencies which are close to 50% and for medium speed engines are approaching 45%.

3.2.1 Engine Characteristics

Tables 3.1 to 3.3 show the main particulars of the slow speed and medium speed engines presently offered by manufacturers. Table 3.1 shows the particulars for the MAN-B&W engines, Table 3.2 the Sulzer engines, and Table 3.3 shows the data for various medium speed engines manufactured by Wartsila, Pielstick, Sulzer, MAN-B&W and MAK.

The need for shipowners to reduce their operating costs has resulted in demands upon diesel engine manufacturers to improve their product. The most important improvements have taken the form:

- reduction of the specific fuel consumption of the diesel engine
- increase of specific output of engines, i.e., an increase of power to weight ratio
- the development of all classes of diesel engines to burn the worst of fuel oils currently available
- design improvements to engines to reduce time required for repairs and maintenance and an increase of maintenance intervals
- the development of self monitoring capabilities of engines in order that maintenance and repair downtime may be reduced and be carried out in the most effective manner
- reduction of manpower required to carry out essential maintenance and operating functions
- world wide standardization of parts between licensees of the engine design.

All of the above contribute directly or indirectly to energy efficiency; tend to be inter-related and have resulted in modifications to the engine design. The following sections discuss the most important aspects of diesel engine design, installation, operations and economy.

3.2.2 Fuel Consumption

The methods used to improve the specific fuel consumption of diesel engines fall into two broad categories:

- improvements to the engine scavenging air system, described in a) through f), and
- improvements to the engine fuel system, described in g) through l)

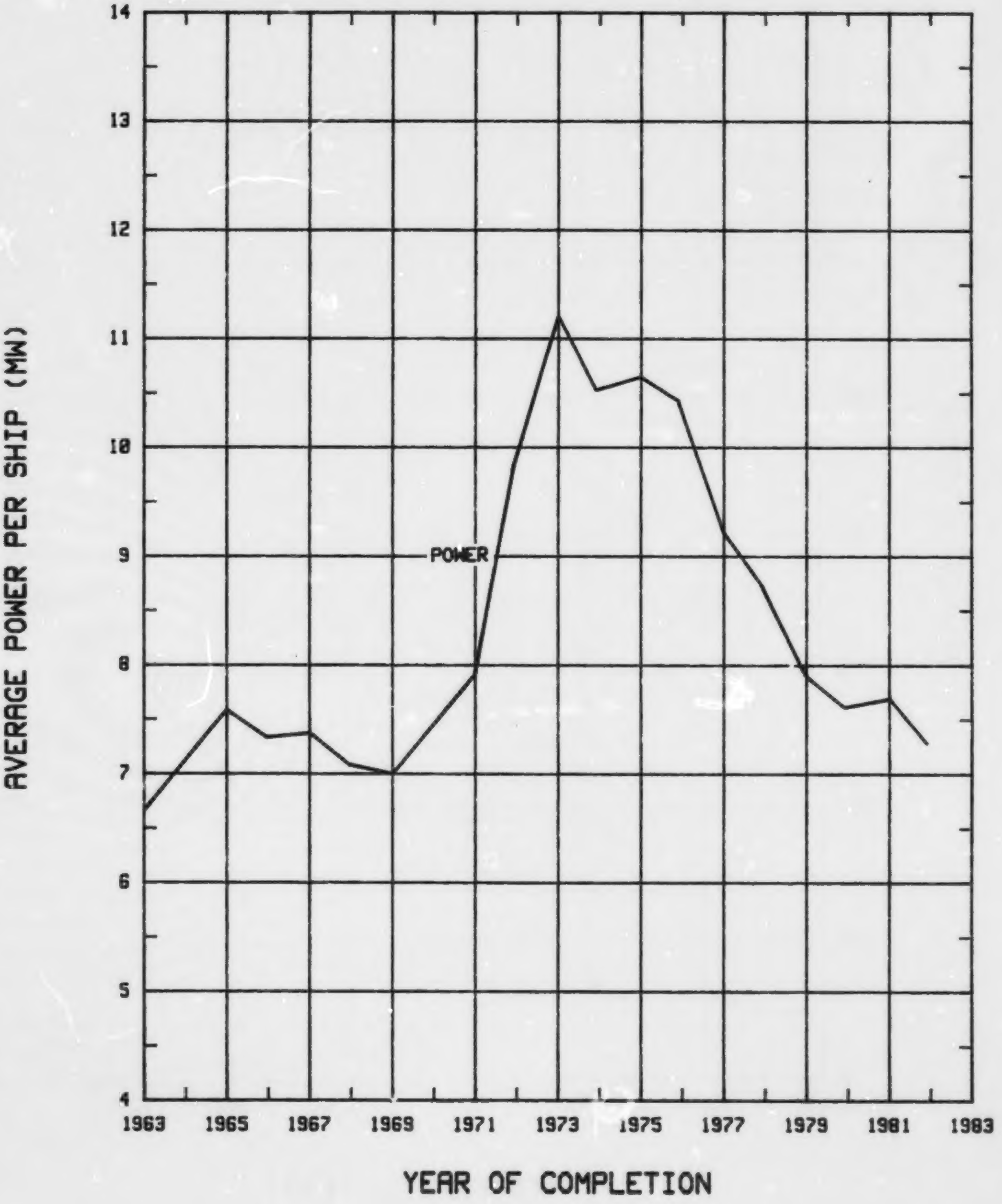
a) Constant Pressure Turbo-Charging

Originally, exhaust gas driven turbo chargers were of the "constant pressure" type but due to design limitations of the turbine and blower, the power output of the engine was limited. The demand for higher power and engine flexibility led to the development of the "pressure pulse" turbo charger system. The pressure pulse system was in common use by many manufacturers until early/mid 1970's when in an effort to improve efficiency a greatly improved constant pressure system of turbo charging the engine was reintroduced. Most large propulsion engines now use the constant pressure system to turbo charge their engines.

This kind of turbo charging has proved to be ideal for long term low load operation on heavy fuel oil and obviates combustion chamber and the scavenge air ducts fouling. The deposits which do occur can be easily removed by water washing. While the constant pressure system is suitable for pressure ratios above 1.2, below 1.2 the pulse system is more efficient. For

FIGURE 3.1

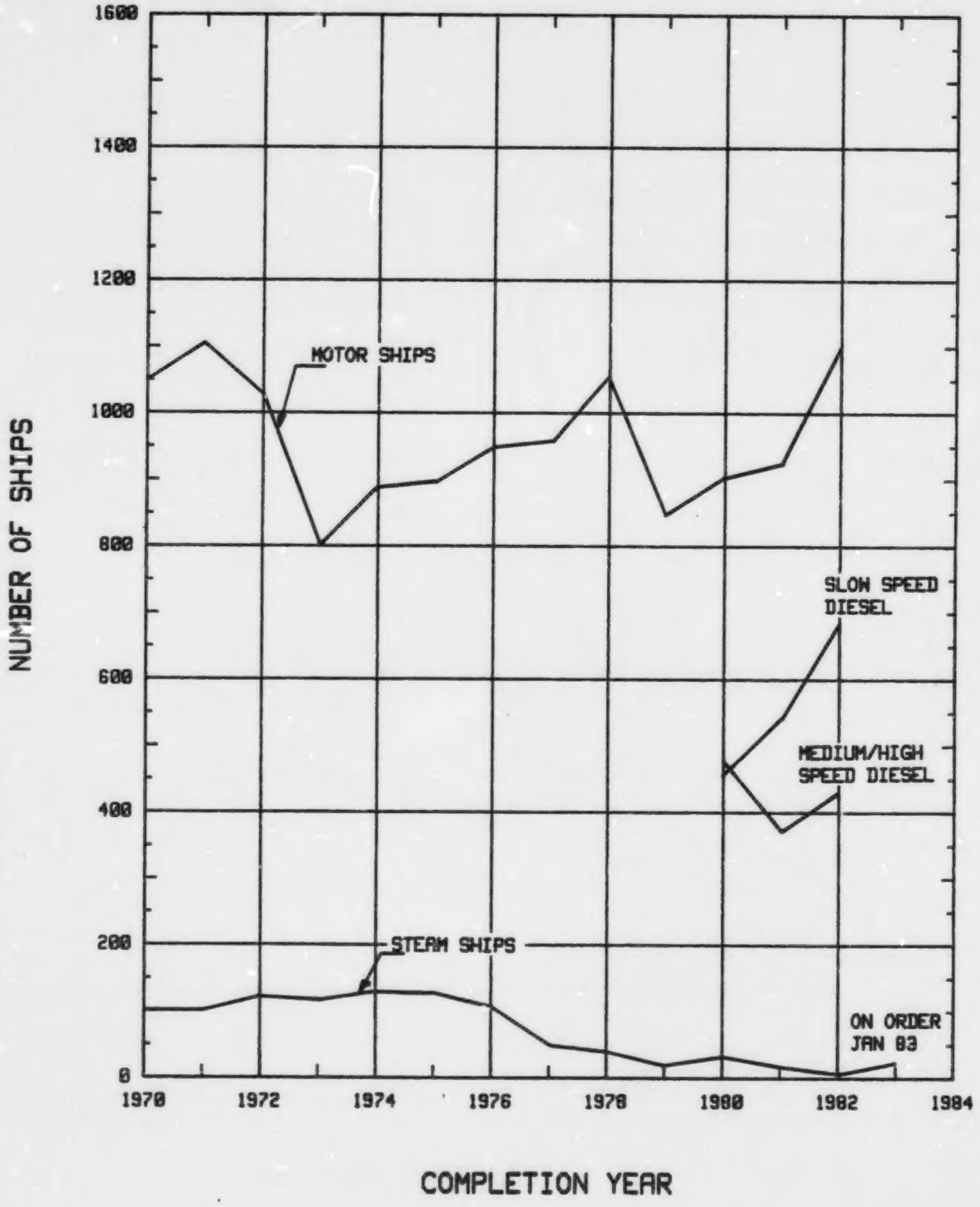
AVERAGE POWER OF SHIPS COMPLETED FROM 1963 TO 1982



[24]

FIGURE 3.2

SHIPS COMPLETED 1970-1982
BY MACHINERY TYPE



[24]

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TABLE 3.1

SLOW SPEED DIESEL ENGINE DATA
M.A.N. - B & W ENGINES

TYPE	BORE	STROKE	NO. OF CYLINDERS	POWER PER CYLINDER	R.P.M.	SFC
	mm	mm	-	kW	-	gms/kW/hr
KGFCA	900	1,800	4 - 12	2,900	117	190
	800	1,600	4 - 12	2,250	130	192
	670	1,400	4 - 12	1,600	150	193
	450	900	4 - 12	725	234	194
LGFCA	900	2,180	4 - 12	2,900	97	185
	800	1,950	4 - 12	2,250	106	186
	670	1,700	4 - 12	1,600	123	188
	550	1,380	4 - 12	1,100	155	185
	450	1,200	4 - 12	725	175	189
LGB	900	2,180	4 - 12	3,360	97	178
	800	1,950	4 - 12	2,600	106	179
	670	1,700	4 - 12	1,840	123	179
	550	1,380	4 - 12	1,190	155	184
	450	1,200	4 - 12	780	175	185
LMC	900	2,925	4 - 12	3,440	74	171
	800	2,600	4 - 12	2,710	83	173
	700	2,275	4 - 8	2,080	95	173
	600	1,950	4 - 8	1,530	111	174
	500	1,625	4 - 8	1,060	133	175.5
	350	1,050	4 - 9	500	200	179.5

N.B. Data taken from latest manufacturer's catalogues. Power per cylinder, r.p.m. and SFC values refer to MCR at ISO conditions and fuel of LHV 42707 kJ/kg. Add 3% to SFC rates for manufacturer's guarantee rates.

TABLE 3.2

SLOW SPEED DIESEL ENGINE DATA
SULZER ENGINES

TYPE	BORE	STROKE	NO. OF CYLINDERS	POWER PER CYLINDER	R.P.M.	SFC
	mm	mm	-	kW	-	gms/kW/hr
RLB	900	1,900	4 - 10,12	2,940	102	181
	760	1,600	4 - 9	2,120	120	182
	660	1,400	4 - 8	1,600	140	184
	560	1,150	4 - 8	1,100	170	184
RTA	840	2,400	4 - 10,12	2,960	87	173
	760	2,200	4 - 10,12	2,420	95	173
	680	2,000	4 - 8	1,950	105	174
	580	1,700	4 - 9	1,410	123	175
	480	1,400	4 - 9	970	150	178
	380	1,100	4 - 9	610	190	181

N.B. Data taken from latest manufacturer's catalogue. Power per cylinder, r.p.m., SFC figures refer to MCR at ISO conditions and fuel of LCV 42707 kJ/kg. Add 3% to SFC figures for manufacturer's guarantee rate.

TABLE 3.3

MEDIUM SPEED DIESEL ENGINE DATA

MANUFACTURER	TYPE	BORE/STROKE	NO. OF CYLINDERS	POWER PER CYLINDER	R.P.M.	SFC
	-	mm mm	-	kW	-	gms/kW/hr
Wartsila	Vasa 32	320/350	4 - 18	335	720	204
Pielstick	PC2-5	400/460	6 - 18	478	520	199
	PC2-6	400/460	6 - 18	550	520	190
	PC3	480/520	6 - 18	700	470	202
	PC4	570/620	6 - 18	1,104	400	191
	PC4-2	570/620	6 - 18	1,215	400	185
Sulzer	AT25	250/300	5 - 18	200	720	204-205
	AS25	250/300	5 - 18	145	750	204-205
	ZA40	400/480	6 - 16	640	560	195-196
	Z40	400/480	6 - 16	550	560	201-203
MAN - B&W		320/360	6 - 18	355	720	195-198 (184-187)
		400/450	6 - 18	550	600	188-192 (178-181)
		400/450A	6 - 18	438	428	
		520/520	6 - 9	885	500	
		520/550A	6 - 18	740	428	186-190 (175)
		580/640	6 - 9	1,135	400	(170-174)
MAK	M332	240/330	6, 8	161	750	203
	M453	320/420	6 - 16	250	600	192-198
	M35	350/380	6, 8	415	750	195-197
	M552	450/520	6 - 16	560-610	500	191-196
	M601	580/600	6 - 9	1,000	425	189-193

N.B. Data taken from latest manufacturer's catalogues. Power per cylinder, r.p.m. SFC values refer to MCR at ISO conditions and fuel of LCV 42000 kJ/kg. MAN SFC values in brackets refer to projected 1984 consumptions. Add 3% to SFC consumption for manufacturer's guarantee rates.

operation at very low engine loads and for start-up purposes the constant pressure turbo charged engine must be fitted with an electrically driven auxiliary blower.

Some shipyards offer retrofit from pulse pressure systems to constant pressure systems by fitting one or two exhaust gas receivers of considerable volume in place of exhaust manifolds [25].

b) High Efficiency Turbo Chargers

In single stage turbo charging the mean effective pressure is closely associated with the charge air pressure that can be achieved. In particular, in the case of slow speed 2 stroke engines it is the turbo-charger efficiency which appreciably influences specific fuel consumption. Specific output and specific fuel consumption therefore depends to a large extent on the development level of the turbo-charger. Some innovations in this field have been introduced lately.

- Turbo-chargers with uncooled gas casings have been designed by BBC and Mitsubishi. The advantages claimed are; higher outlet temperatures (10-15°C) favouring steam production in exhaust boilers and generally higher performance due to reduced loss of energy through cooling [26].

- Variable pitch turbine nozzle ring developed by MAN is used to rotate the nozzle blades when the gas flow is low so as to close partially the area of the nozzles and thus increase the turbine speed to maintain the charge pressure. This greatly improves turbo-charger performance at low loads [26].

- Backward curved impeller blades for the compressor have been introduced by Mitsubishi in order to raise the total efficiency of the turbo-charger. A comparison of the compressor with backward curved blade impeller and the straight radial blade impeller shows that with the former the surge line is located away from the point of compressor peak efficiency. Thus, when matched with the engine working line showing, say 15% as a margin of safety for surging avoidance, the compressor with the backward blade impeller exceeds the compressor with the straight radial blades by 2-4% [27].

- Two stage turbo-charger provided with charge air cooling after each stage of compression has been applied by MAN. This solution offers greater efficiency than single stage turbo charging, even at a given charge air pressure [28].

The new designs of turbo-chargers with uncooled gas casings do not suffer from acid corrosion and due to the unlimited in service life of the plain bearings, require no scheduled maintenance for 4 years. Therefore the most important aspect of the maintenance is the removal of fouling from the turbine section of the turbo-charger. Cleaning of the turbine during operation can be performed most readily by the dry cleaning method in which a cleaning medium such as cereal grain is fed into the turbo-charger from its forward end. An alternative is the hitherto prevalent water cleaning method. Both methods require fitting of the adequate feeding devices. Cleaning the turbines is extremely important to maintain efficiency when the engine burns residual fuel oil.

c) Uniflow Scavenging/Exhaust System

Recently, Sulzer and Mitsubishi developed a new computer analytical method for the research of scavenging gas flow in a cylinder. As a result the most suitable scavenging arrangement could be selected. It became evident that scavenging efficiency of the uniflow system is better than that in the loop scavenging, especially if the stroke/bore ratio becomes larger. The main advantages of the uniflow system are; low flow resistance, low heat load and effective fuel distribution in the fresh air charge [27].

d) Reduction in Valve Overlaps

This is another feature of scavenging which improves engine efficiency. By delaying the opening of the exhaust valve the effective piston stroke is increased and the heat losses due to the air charge renewal are less.

e) Exhaust Gas Diffusers

Some engine types have diffusers fitted at the exhausting outlet. Because of the inertia effect of the exhaust gas due to these diffusers, a considerable decrease of pressure takes place in the cylinder when the scavenging port opens after blow down, which gives improvement of scavenging.

f) Exhaust Valves

The exhaust valve has been a major source of problems in the past, more especially with medium speed four stroke engines when a heavy grade of fuel oil is used. The valves are prone to damage due to hot corrosion brought about by the deposition of compounds of vanadium and sodium on the valve seat. Complex salts of vanadium and sodium will tend to deposit on the exhaust valve seats if the exhaust gas temperature is above the melting point of the salts. When the deposit thickness increases, a piece will eventually break away, forming a gas channel that will quickly lead to valve burning. Hot corrosion occurs when the valve cone temperature exceeds about 550°C, and is often the decisive factor in the valve's life. This problem has become of increasing importance in recent years as the four stroke medium speed engine is developed to burn residual fuels and the specific output per cylinder has increased. To improve the exhaust valve resistance to damage of this type, nimonic steel and ceramic materials and coatings are being used and great attention given to valve design as follows:

- Cooling of the valve seat with water by means of bores and/or annular passages. There are also attempts to introduce water, or oil, cooled valve spindles.

- A small angular difference between the valve seat and lid in the cold condition provides a wider valve seat contact area when the valve is hot and exposed to the combustion pressure. This wider contact will further ensure an efficient heat transfer from the valve spindle to the bottom piece [29].

- Modification of the gas duct to improve the gas flow.
- Provision of the valve with a separate valve cage to facilitate replacement.
- Rotation of the valve spindle by the exhaust gases. The speed of rotation is about 1-2 r.p.m. The rotating valve ensures a more uniform seat temperature and a continuous levelling of seat indentations to reduce the risk of burning.
- Elimination of mechanical springs by use of the pneumatic spring.
- The use of hydraulic pushrods controlled from a valve actuator on the camshaft to provide more accurate valve opening.

g) Higher Injection Pressure

Due to modern refinery processes and the demand for gasoline fuels, the aromatic content of residual fuel is increasing. Fuel with a high aromatic content will cause ignition lag and detonation particularly at low load. Detonation due to ignition lag brought about by the aromatic content of the fuel can be overcome by:

- a hot combustion chamber achieved by higher and constant pressure turbo charging
- higher compression ratio
- high injection pressure

The last factor is the most important. It has been proven that injection pressure of 1,000-1,200 bar gives fuel droplets of 20 micron diameter and aromatic chains broken to such small pieces burn easily without delay. Injection is particularly efficient if the rise and cut off time of the injection is rapid. This is achieved, for example, by MAN by use of a pressure equalising system in the injection pump.

h) Heating Arrangement

Engine fuel injection systems are designed for fuel viscosity of about 14 cSt. Heavy fuel must therefore be heated and in order that the heat is not dissipated in the pumps, pipes and injectors at low load, the fuel must be circulated through the system. Fuel is commonly preheated to 120°C and as fuel quality worsens and the viscosity increases this temperature will increase.

It is advantageous if all fuel carrying components are located in a "hot box" which reduces the heat loss from the fuel and the preheating time at cold start.

i) Fuel Valves

Due to the different composition of the products of combustion between heavy and light fuel, the engine runs hotter when heavy fuel is burnt. The molecules of hydrocarbons are larger in heavy fuel than in distillate fuel and burn more slowly. As the larger molecules burn they tend to migrate more closely to the walls of the combustion chamber and more heat is transferred to the valves, cylinder head and piston crown than when distillate fuel is burnt.

As light fuel has a higher hydrogen content than residual fuel, more H_2O is formed during combusstion which has the affect of increasing the heat transfer coefficient of the exhaust gases. Hence more heat is transferred from the gases to the cylinder walls during the expansion stroke and when light fuel is burnt the combustion chamber metal temperatures and the exhaust gas temperature are cooler than when heavy fuel is consumed.

High gas temperatures in the combustion chamber will tend to disrupt the fuel valve function unless the valve is especially cooled. The shape of the fuel spray from the atomizer into the combustion chamber is very important and unless the atomizer is maintained at peak efficiency the thermal efficiency of the engine will quickly fall.

An interesting valve design with automatic control of fuel valve opening pressure has been presented by Mitsubishi in the UEC type engine. The pressure control automatically turns on or off the control air of the actuator located at the top of the valve. The pressure of the needle valve spring can thus be varied according to the engine load.

3.2.3 Lubrication Oil Consumption

The lubrication oil consumption and cost can be significant for the prime mover system and should be included in any energy cost equation for machinery selection.

The medium speed oil engine is normally a trunk piston engine fitted with an integrated lubrication system for the running gear, cylinder and crankshaft. The oils used in these engines are detergent oils with additives to counter oxidation and corrosion and to neutralize acidity due to the products of combustion. The oil system for the medium speed engine and its associated gearbox are always separate. Oil losses in these engines are due to crankcase leakage and loss of oil through cylinder lubrication.

The cylinder and the running gear lubrication systems of slow speed engines are always separate. In some cases a separate system is also provided for the crankshaft lubrication. Different qualities of oil are used for cylinder lubrication to that used for the crankcase. The cylinder lubrication oil is consumable and has a high percentage of alkaline additives to neutralize the acidic products of combustion and is designed not only to lubricate the piston rings and cylinders but to provide protection against corrosion. The oil used for the running gear and the crankcase is straight run mineral oil similar to that used for the turbine and gearbox systems.

Medium speed engine lub oil consumption is in the range 1 to 1.5 grams per kilowatt hour and for the slow speed engine, crankcase consumption is approximately 0.25 grams per kilowatt hour and between 0.5 and 0.7 grams per kilowatt hour for the cylinder lubrication system. The consumption of oil in the diesel engine is due to leakage and by losses of oil within the combustion spaces. All cylinder lubrication oil is lost and the rate is dependent upon engine speed and power developed by the cylinder.

The cost of lubrication oil is estimated as:

- medium speed oil engines, 4 to 12 U.S. dollars per kilowatt year
- slow speed oil engines, 4 to 8 U.S. dollars per kilowatt year.

A review of the actual lubrication oil costs, based upon 1981 prices, of a prominent operating company, revealed the following:

- medium speed oil engine installations, 8.4 U.S. dollars per kilowatt year
- slow speed oil engine installations, 7 U.S. dollars per kilowatt year.

3.2.4 Maintenance and Repair Costs

As a percentage of variable operating costs, eg. fuel, crew costs, insurance, maintenance, etc., maintenance is not a major factor. Modern marine engines of all types are becoming increasingly more reliable and the costs associated with maintenance and repairs of engines are now a smaller percentage of the total operating costs than before. As the quality of the bunkers is expected to deteriorate, the development of engines to burn the cheapest and poorest quality fuels, without incurring an increased maintenance and repair cost, will continue.

When vanadium and sodium, (vanadium from the crude source and sodium from sea water contamination due to careless storage or transportation), in fuel oils are burnt in either reciprocating machinery or in boilers, vanadium pentoxide is formed giving rise to corrosion of the boiler tubes or the exhaust valves seats in the engine. This type of problem is common and typifies the affect of fuel quality on maintenance.

Scheduled maintenance requirements for diesel engines are usually well defined by the manufacturers and are based upon operating experience with various grades of fuel oil. In general, medium speed oil engines require more maintenance time than the slow speed engines. From research of relevant literature the maintenance and repair costs as a percentage of total operating costs are:

- slow speed diesel engines, 4-5%
- medium speed diesel engines, 4-6%

In recent years the reliability of diesel engines of all types have improved considerably and unscheduled outages required for maintenance now are less. Medium speed diesel engines generally require more and higher expertise of maintenance than slow speed engines, however, the maintenance tasks are often less arduous and occupy less time. In many installations two or more medium speed engines are fitted in parallel to a gearbox thus permitting a large reduction of downtime as one engine can be maintained while the vessel can operate using the remaining healthy engines.

To arrive at any generalised conclusions with respect to downtime due to repair and maintenance may be misleading but evidence does tend to indicate that slow speed oil engines require fewer hours for both planned and unplanned maintenance.

3.2.5 Flexibility and Part Load Operation

Oil engines have a relatively flat SFC-load characteristic with a minimum fuel rate at 75 to 80% of the MCR design conditions. Consequently the specific fuel consumption of these engines may improve when operated under part load conditions. Figure 3.3 gives the average SFC characteristics of medium and slow speed diesel engines.

Continuous service power (CSR) for oil engines has historically been 85% of MCR, but this has tended to increase in recent years to 90 - 93% MCR by a reduction of margins for deterioration of ship performance with time. This may be justified due to the large improvements made in the fouling resistance of hull coatings. In order to achieve maximum fuel economy a fouling margin should be added to the bare hull resistance and to the ships ancillary equipment. Provided the vessel's inservice resistance is known the engine can be specified to provide the required power at the lowest point on the SFC curve.

The flat characteristic of the oil engine allows flexibility of operating speed without the penalty of increasing the rate of fuel consumption. There is a limitation on the minimum operating speed and extended operation at low loads on heavy fuel may cause engine fouling unless the fuel system is modified accordingly. The affects of substandard maintenance and operating procedures during the vessel's life has considerably greater effect on the vessel's fuel consumption than the design SFC. It has been estimated that savings in fuel up to 10% may be achieved in most vessels solely by improvement to maintenance and operating procedures [30].

Shipowners are now aware of the need for good fuel consumption over a range of service speeds. This requirement has given impetus to the ordering of vessels fitted with multiple engine systems. While these systems are more expensive than single engine systems, multiple engines give significant advantages if the ship is required to operate at a wide range of powers. Systems with more than one engine also gives significant advantages with respect to redundancy.

3.2.6 Weight and Volume

An increase in the weight and volume of the machinery installation for a cargo ship often will lead to loss of deadweight or cargo carrying capacity.

The engine output or power per cylinder has increased steadily over the last ten years and the cost per horsepower in real terms has decreased. The increase in engine performance is principally due to improvements of materials, foundry techniques and cooling system designs.

Recent developments in the production of steel alloys, foundry techniques and measurements and analysis of the mechanical and thermal loads on critical components of the engine, have allowed the thermal design of engines to be advanced and permit higher temperatures and pressures to be used. Higher cylinder pressures and temperatures have allowed the power output per cylinder to be increased and efficiency to be improved. The development of a popular model of medium speed four stroke engine manfactured by Stork-Werkspour is shown in Figure 3.4.

Figure 3.5 is derived from manufacturers' catalogue data and shows the typically installed weight of machinery for two propulsion systems in the power range 5 to 30 megawatts. It can be seen from the figure that the slow speed engine installation is heavier than a medium speed engine system for the same power. On the basis of tonnes per megawatt output, long stroke diesel engines are 14 to 20% heavier than earlier short stroke designs.

The size of a ship's engineroom is related to the type and power of the machinery. In the low and medium power range the geared medium speed engine provides the most compact system. Slow speed engines are high and their length is a function of the number of cylinders. The new generation of long stroke slow speed engines may also require a wider engineroom than previous engines as the width of the bed plate has increased to accommodate the larger crank throw.

It is difficult to be specific with regard to minimum engineroom lengths for the two types of propulsion machinery. However, it would be expected that medium speed geared installations will require less space overall due to their higher kilowatt per cubic meter ratio. If there is a limitation on engineroom height, as is often the case with special ships such as Ro-Ro and ferries, then the medium speed engine has a distinct advantage compared to the slow speed engine.

3.2.7 Main Engine Monitoring

Experience indicates that significant fuel savings can be obtained through the judicial use of a good combustion analysis system. Savings in the range of 6 to 12%, while dramatic, are achievable on medium and high speed engines and 2 to 4% savings are realistic for the larger slow speed diesels. The type of instrument which is currently finding favour in many fleets is the MIP (Mean Indicated Pressure) calculator. Many similar MIP indicators are now available and a typical example would be the Autronica NK-4 made by Autronica A/S, Trondheim,Norway. This instrument has many more functions than solely a calibration for the mean indicated pressure in the cylinder and promises to be a very powerful analytical tool.

Operators are able to optimize the timing of their engines due to the precise information provided by the combustion analysis units. It is estimated that for each 1° that combustion time is retarded, there is approximately a 2% increase in the specific fuel consumption. Medium and high speed diesel engines often operate with 3 - 5° retardation from the optimum condition. Accurate timing cannot be achieved by the present technique of indicator diagrams which have, under the best of circumstances, a 7 - 10% accuracy and which are quite useless for medium and high speed engines.

FIGURE 3.3

SPECIFIC FUEL CONSUMPTION
VS ENGINE LOAD

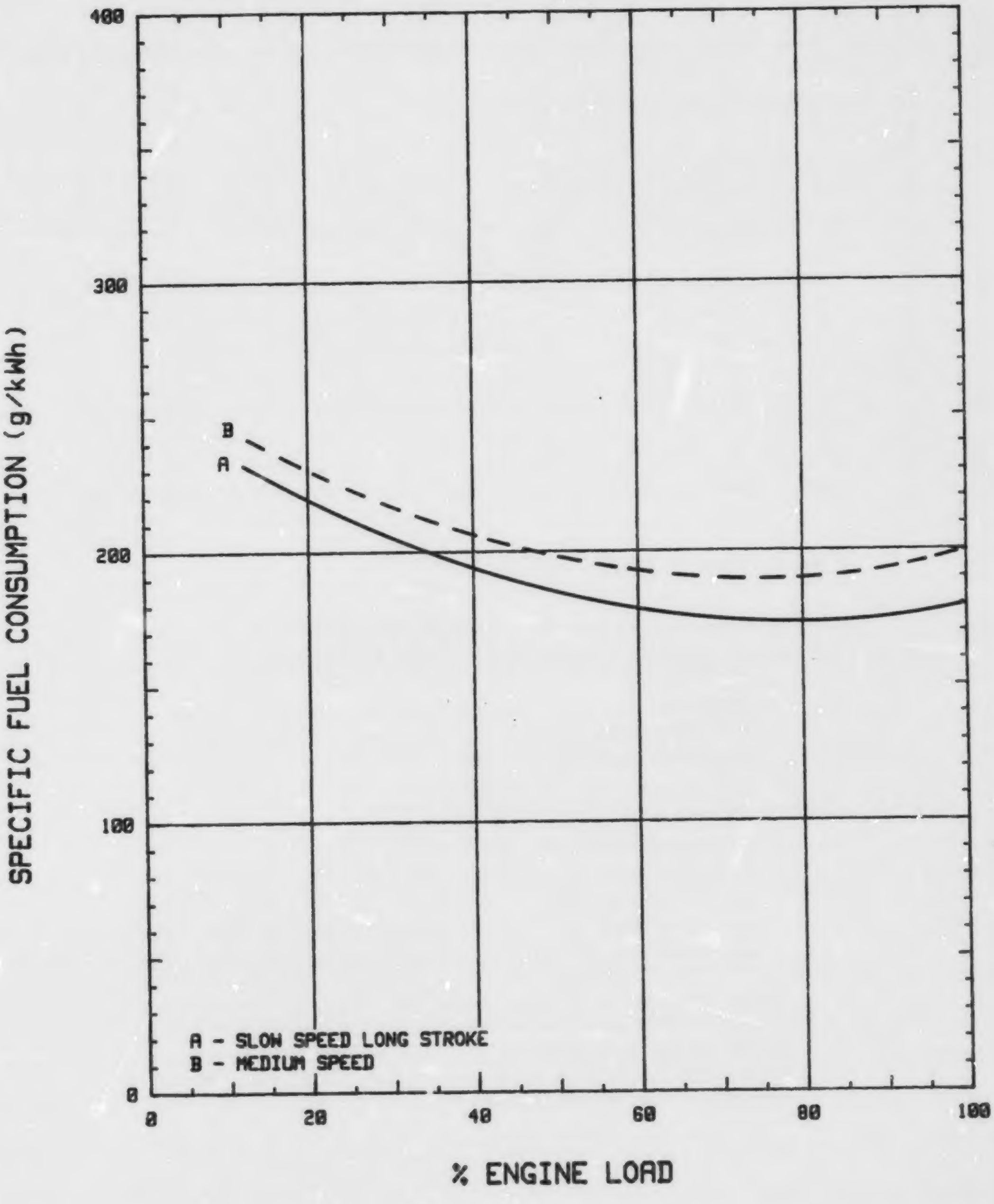

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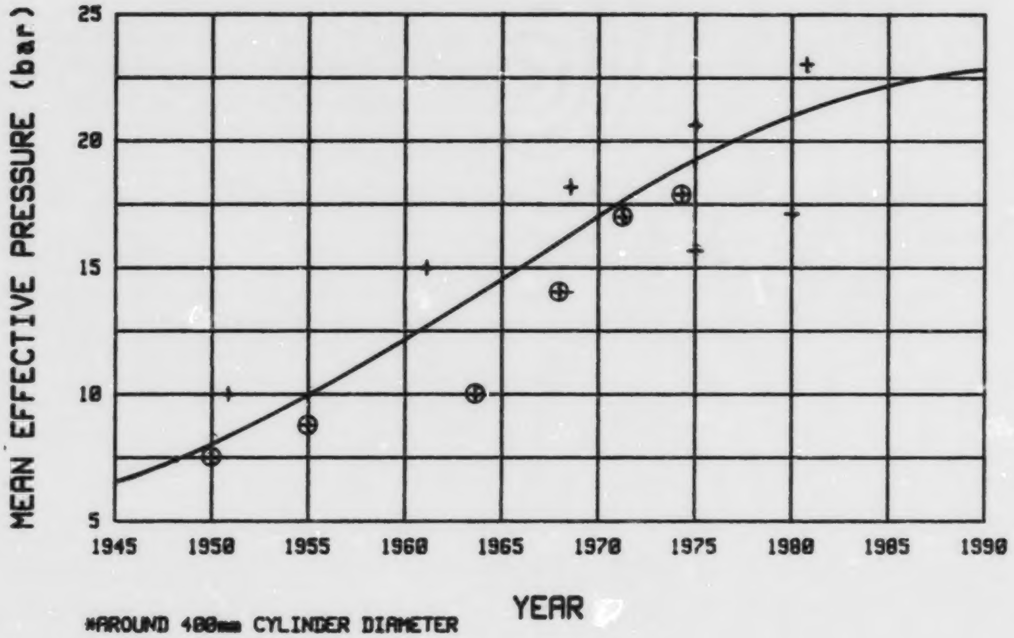
FIGURE 3.4

TREND OF MEAN EFFECTIVE PRESSURE
OF MEDIUM SPEED ENGINES



*AROUND 400mm CYLINDER DIAMETER

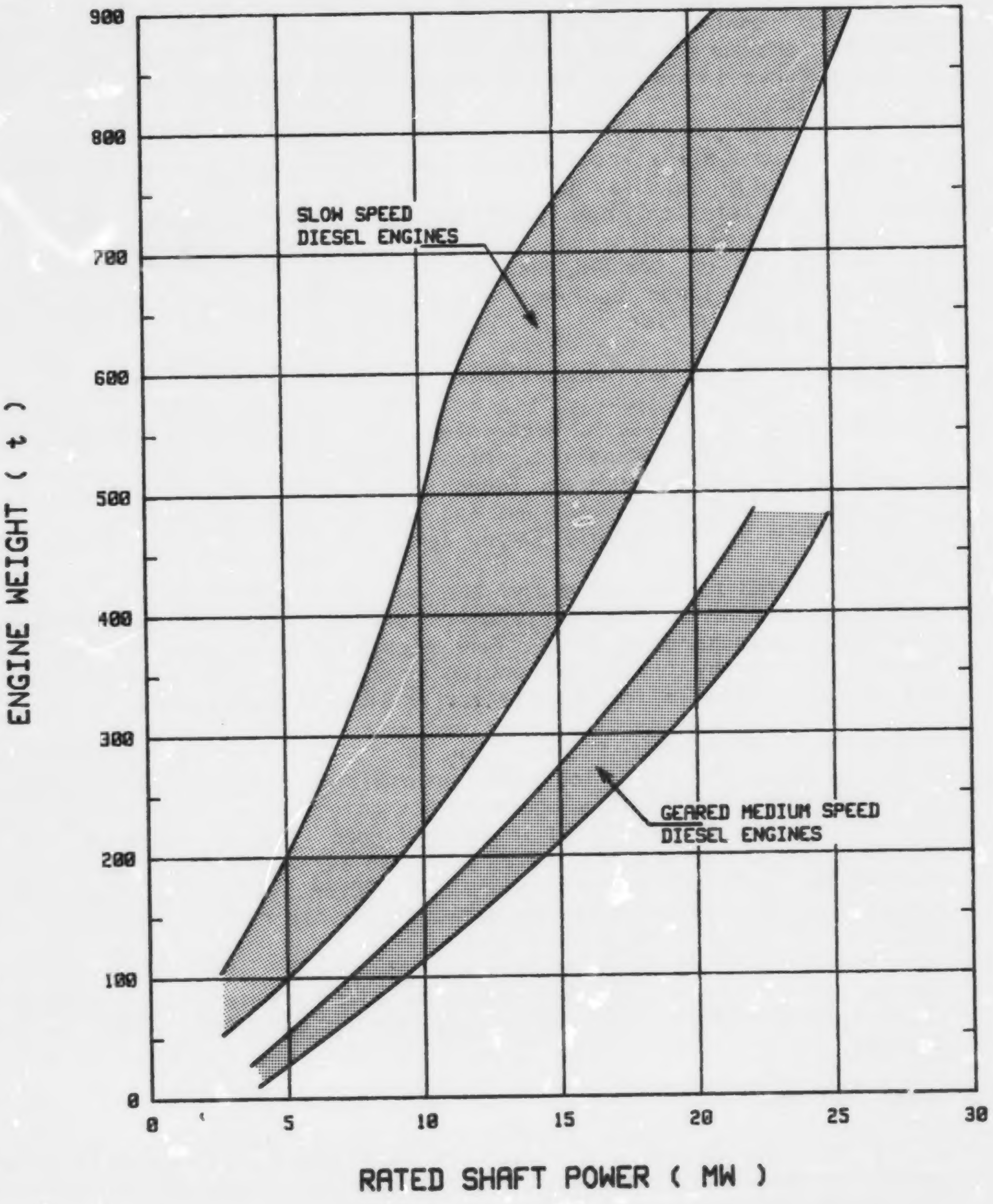
YEAR		1968	1971	1974	1982	19...
RATING/CYLINDER	kW	370	490	550	565	565 - 630
ENGINE SPEED	1/MIN	500	550	600	600	600
MAIN EFFECTIVE PRESSURE	BAR	14,3	17,3	17,7	18,2	18,2 - 20,3
MAX. COMBUSTION PRESSURE	BAR	95	115	120	130	145 - 160
BOOST PRESSURE	BAR	2,3	2,7	2,8	3,0	3,1 - 3,5
AIR/POWER RATIO	KG/kWhr	8,7	8,4	8,4	7,5	8 - 7,5
SPECIFIC FUEL CONSUMPTION	gr/kWhr	216	212	203	192	180 - 185
EXHAUST GAS TEMPERATURE	°C	390	405	410	410	400 - 410
MAX. CYLINDER HEAD TEMP.	°C	265	265	250	250	225 - 250
MAX. PISTON TEMPERATURE	°C	305	270	300	300	275 - 300
MAX.EXHAUST VALVE SEAT TEMP.	°C	440	450	450	450	425 - 450

Above. Progress in medium speed engine design typified by SW Diesel's TM410 series development. Note the substantial increases in specific rating from 1968 until 1974 and the decrease of specific fuel consumption from 1974 until 1982. Future design rating trends are indicated on the right of the table

[MOTOR SHIP, JAN. 1984]

FIGURE 3.5

ENGINE WEIGHT COMPARISON



ENGINE MASSES WITHOUT OIL/WATER

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The modern diesel engine combustion monitoring units utilize the latest designs in integrated microelectronics to provide readily understandable video displays of the complete engine combustion process. Permanent records may be obtained through a printer or may be taken from the oscilloscope by polaroid camera.

The MIP unit receives signals, from transducers fitted to the engine, which are proportional to scavenge pressure, crank angle, cylinder pressure and fuel oil pressure before injection. Figure 3.6 shows a sample cylinder pressure curve in various magnifications and Figure 3.7 shows fuel pressure curve [92].

Microprocessors allow provision of complete diesel analysis systems. Data from the digital display unit is continuously transmitted to the control room terminal station which performs the trendline analysis and stores the information on magnetic tape. All of the information is available on demand and trendlines or comparisons are generated automatically on the video display. Such trend analyses are very effective methods for detecting engine problems in time.

The range of diesel engine monitoring systems includes:

- MEGASYSTEMS analysers, Megasystems Inc., Boca Raton, Florida, U.S.A. (in Canada APE Canada Inc., Mississauga)
- CYLDET by ASEA, Vasteras, Sweden
- McDOT by A/S Norsk Electrisk and Brown Boveri, Oslo, Norway
- COMOS-D4 by Mitsubishi, Tokyo, Japan

Closely related to the MIP system are various engine diagnostic systems that continuously control the condition of piston rings, cylinder liners, main bearings, turbochargers, etc. This enables the operator to optimize for full efficiency by timing his engine precisely to the fuel characteristics and in addition helps to identify faults in vital components.

The Sulzer Integrated Piston Ring Wear Detecting Arrangement (SPIWA) involves using a special top piston ring incorporating a wedge shaped band of non magnetic material, whose width is detected as a measure of ring wear by an inductive sensor fitted in the cylinder liner at scavenge port level. The wear pattern of the top piston ring can be automatically monitored while the engine is running. Furthermore, the SPIWA system indicates the actual rotation of the piston rings and detects possible ring sticking and breakage [93].

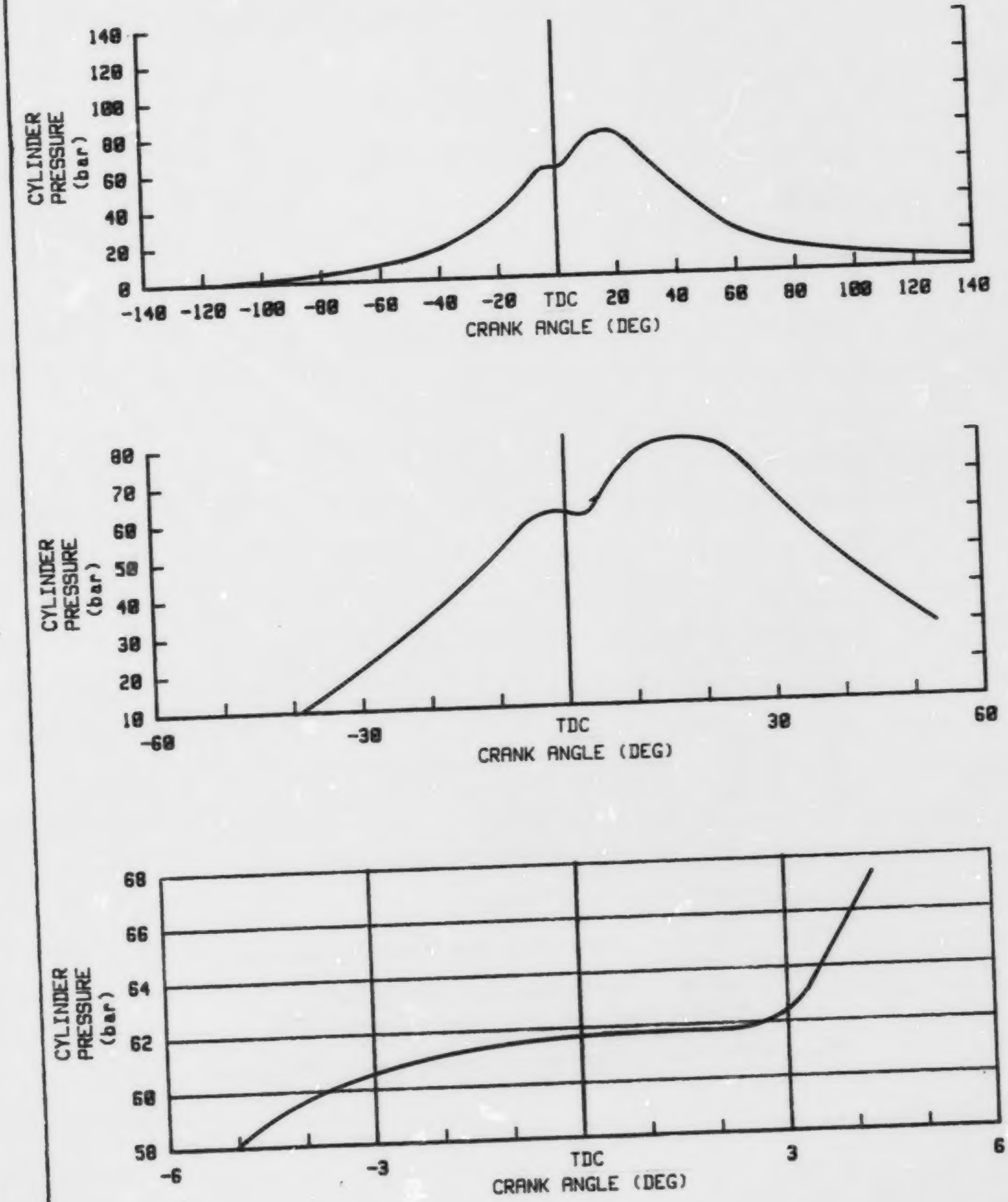
The output of the sensors are fed to a microcomputer which processes the signals to provide a read out in engineering data from which wear can be deduced.

3.3 Steam Turbines

Up to the time of the oil crisis in 1973 steam turbines were a popular choice by shipowners for vessels with power in excess of about 7,500 kW. The steam turbine found favour in tank ships and other vessels where large auxiliary power was required in port for pumping cargo and the steam for the boilers could be utilized for this purpose.

FIGURE 3.6

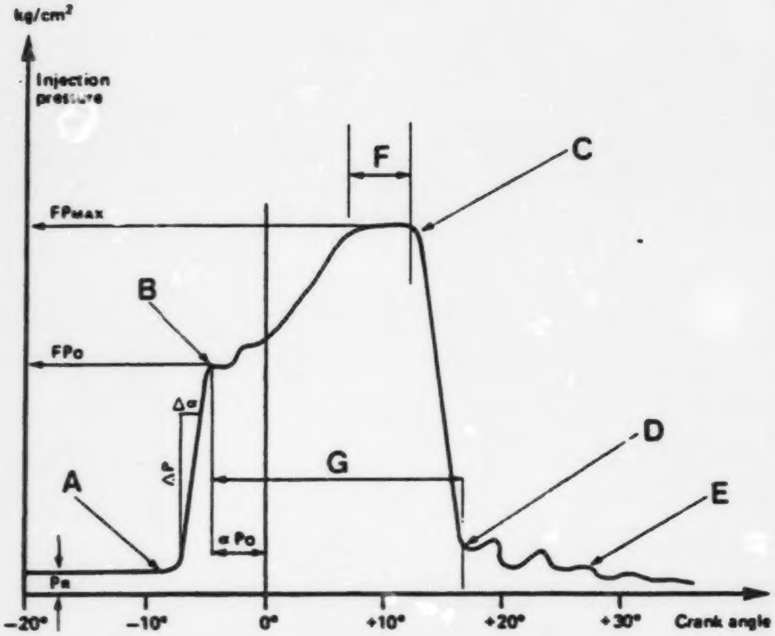
SAMPLE OF DISPLAYED CYLINDER PRESSURE CURVE IN VARIOUS MAGNIFICATIONS



[92]

FIGURE 3.7

TYPICAL FUEL PRESSURE TRACE AT APPROX FULL POWER



A – Pump spill closes; 8° approx BTDC
B – Fuel valve opens; 4° approx BTDC
C – Spill opens; 12° approx ATDC
D – Fuel valve closes; 16° approx ATDC
E – Reflected pressure wave due to fuel valve closing
F – Partial equilibrium
G – Injection period; 20° approx
P_R – Residual fuel pressure
$\Delta P/\angle\alpha$ – Rate of fuel pressure rise before fuel valve opens; 2·5 approx
FP_O – Fuel valve opening pressure; 350 kg/cm²
P_O – Angle at which fuel valve opens relative to TDC; 4° approx BTDC
FP_{MAX} – Maximum fuel pump discharge pressure; 650 kg/cm²

Since 1973 the oil engine has replaced the steam turbine for powers up to 37,500 kW chiefly due to the oil engine's better fuel consumption.

3.3.1 Fuel Consumption

Figure 3.8 shows the all purpose design fuel rate in grams per kilowatt hour for steam turbine systems at various horsepower ratings [34]. The figure shows a range of fuel rates depending upon the degree of sophistication of the steam cycle and the boiler steam conditions selected. The fuel rates shown in the figure refer to present and projected practice at design conditions and does not refer to part load conditions. Curve A shown in this figure, will be applicable to virtually all modern steam plants now in operation at sea. Although manufacturers have attempted to promote reheat steam turbine systems for marine use, to improve the fuel consumption, very few vessels have been built with this machinery. Curves B, C and D represent the projections of the steam turbine industry for the future more sophisticated vessels. It is believed that shipowners are unlikely to fit sophisticated steam turbine systems with multi stage feed heating and reheat cycles.

Although there are no technical reasons for not using high temperature high pressure reheat steam systems at sea, the increased capital cost of the equipment, the sophistication and possible adverse effects on system availability will probably not be compensated for by an overall decrease in operating costs. In addition to this fundamental objection to an increase in the sophistication of marine power plant systems, there is a worldwide decrease in the number and quality of engineers who are willing to go to sea. This lack of suitably qualified operating personnel compounds the problems associated with sophisticated plants.

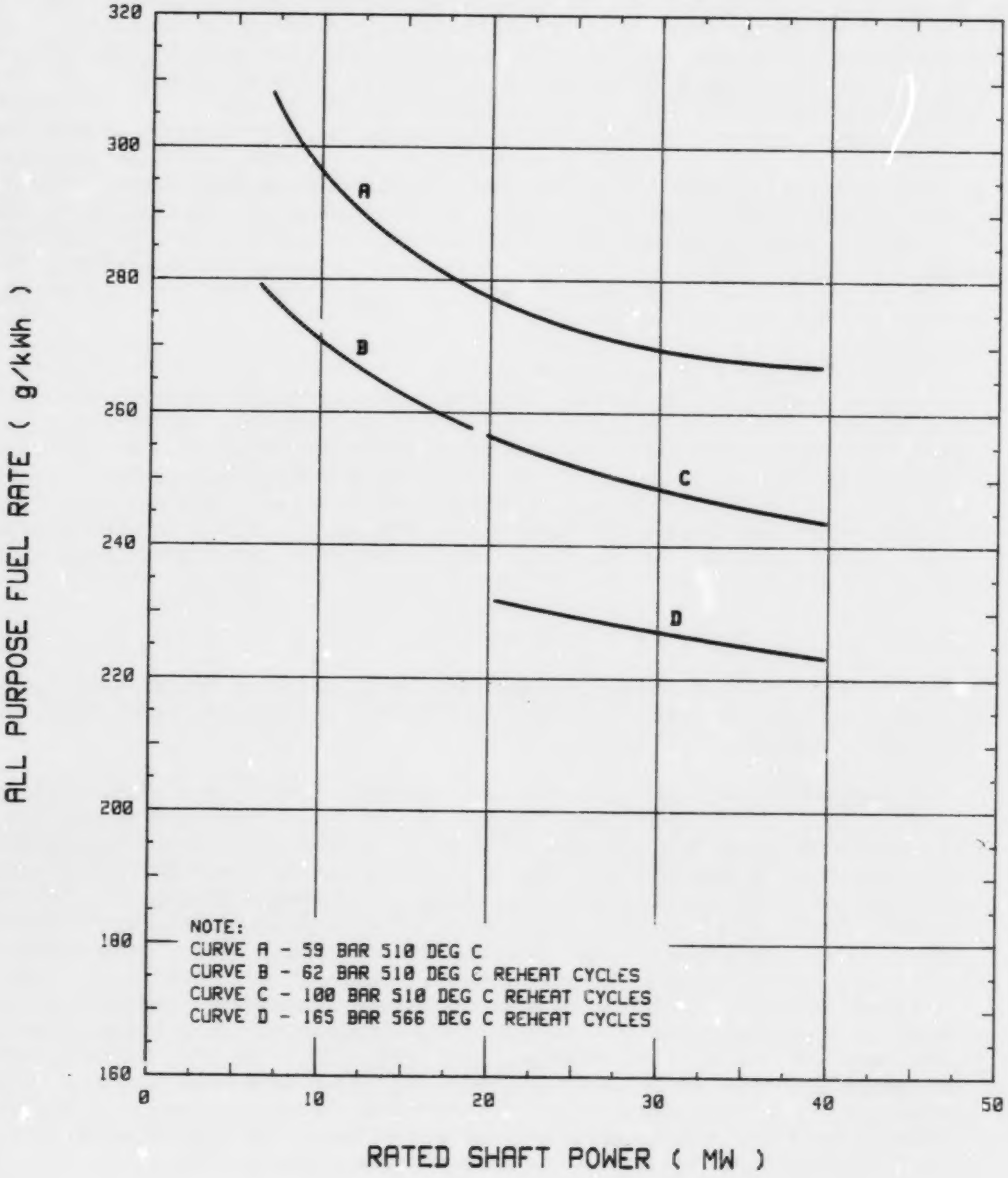
3.3.2 Flexibility and Part Load Operation

The marine steam turbine system comprising boilers, steam turbine, feed heaters, condenser and pumps is an extremely inflexible power plant if the system is designed for maximum efficiency.

Turbines which are designed for maximum efficiency invariably have small clearances and are therefore subject to damage if the turbine casings are àsymmetrically heated. Damage may also be caused by differential expansions between the rotors and the casing if startup is too rapid. Therefore, changes of turbine load causing changing of temperatures of the casing either during start-up or reduction of load must be carried out slowly in order that the casing may restabilize at their new temperature. In addition, feed heaters in the system, supplied by steam bled from the turbines are only suitable to operate between relatively narrow limits and must be isolated when the turbines are maneouvred. The boilers can have problems if rapid load changes are required. Modern boilers contain relatively little water and the steam and water drums are usually of small diameter. During manoeuvring, when the steam to water ratio in the boiler changes quickly, large variations of water level can result which may lead to tripping of the boiler in the most extreme circumstances.

FIGURE 3.8

ALL PURPOSE FUEL RATES FOR STEAM TURBINE POWER PLANTS



[34]

Steam turbines which are directly coupled to a propeller can be flexible, i.e., have rapid manoeuvring characteristics but this is achieved at the expense of turbine efficiency. Such systems are unlikely to be selected for the future energy efficient ship.

Steam turbine installations operate well at part load although the efficiency of the system suffers dramatically. Unlike electrical generating plant ashore, marine steam turbines are variable speed machines which are coupled to fixed pitch propeller systems, variable frequency alternators or variable speed d.c. electrical generators. Therefore part load operation is achieved by variation of the propeller speed and hence the turbine speed. The efficiency of an impulse turbine is principally influenced by the ratio of the turbine blade speed to the steam speed leaving the turbine nozzle. And the optimum blade to steam speed ratio for maximum efficiency is 0.5. This point of maximum efficiency is achievable at the design point of the machine, at part load the efficiency of the turbine reduces considerably.

The reduction in turbine efficiency or total plant efficiency is the summation of the change of efficiency of the various components within that plant. A reduction of load may increase the boiler efficiency although turbine efficiency will decrease. Similarly a decrease in load will reduce the quantity and quality of the steam available from the turbine for feed water heating and thus require the shortfall of feed heating steam bled from the turbine to be supplemented by live steam from the boiler. This will of course increase the specific fuel consumption of the system. Figure 3.9 shows the specific fuel consumption of a turbine plant at part load, and is typical for any marine steam turbine installation. Figure 3.10 shows the change in efficiency of a marine boiler with load [33].

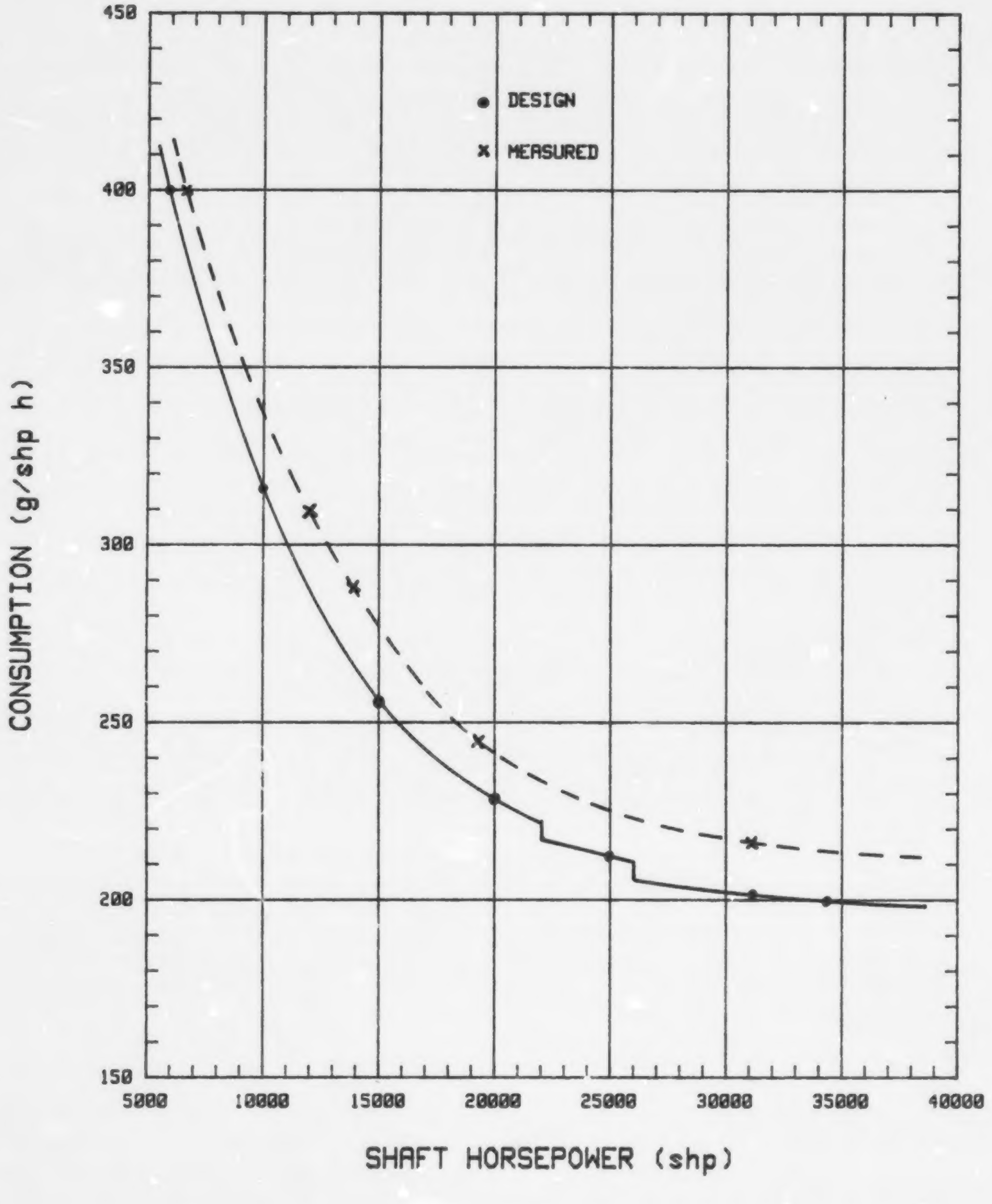
Part load operation of steam turbines can only be improved by fundamental changes to the system. The adjustments which are required generally take the form of changes to the nozzle arrangements of the inlet stages of the HP turbine and changes of the HP turbine gear ratio to allow the turbine to operate at its designed speed at reduced propeller speed. In addition the feed heating arrangements must be changed to suit the change of steam pressures at the bled points in the turbine. This modification may be most economically made by cascading steam from a high pressure extraction to a lower pressure point in the cycle. If adequate steam pressures can not be maintained in the original turbine extraction stage, at reduced power, steam may be supplied from an upstream bleed point through a cascade valve set to hold a preset minimum pressure. This system may be arranged to allow the high pressure and low pressure heaters to sustain performance across a range of low power operation. A typical Cascade system is shown in Figure 3.11 [31, 32].

3.3.3 Turbine Monitoring

One of the most crucial components of steam plant efficiency is the control of combustion air flow to the furnace. The air registers and oil burners of modern boilers are designed to minimize the uptake heat loss. This is achieved by good atomisation by the burner and accurate control of primary and secondary air by the register. In order for the optimum air fuel ratio

FIGURE 3.9

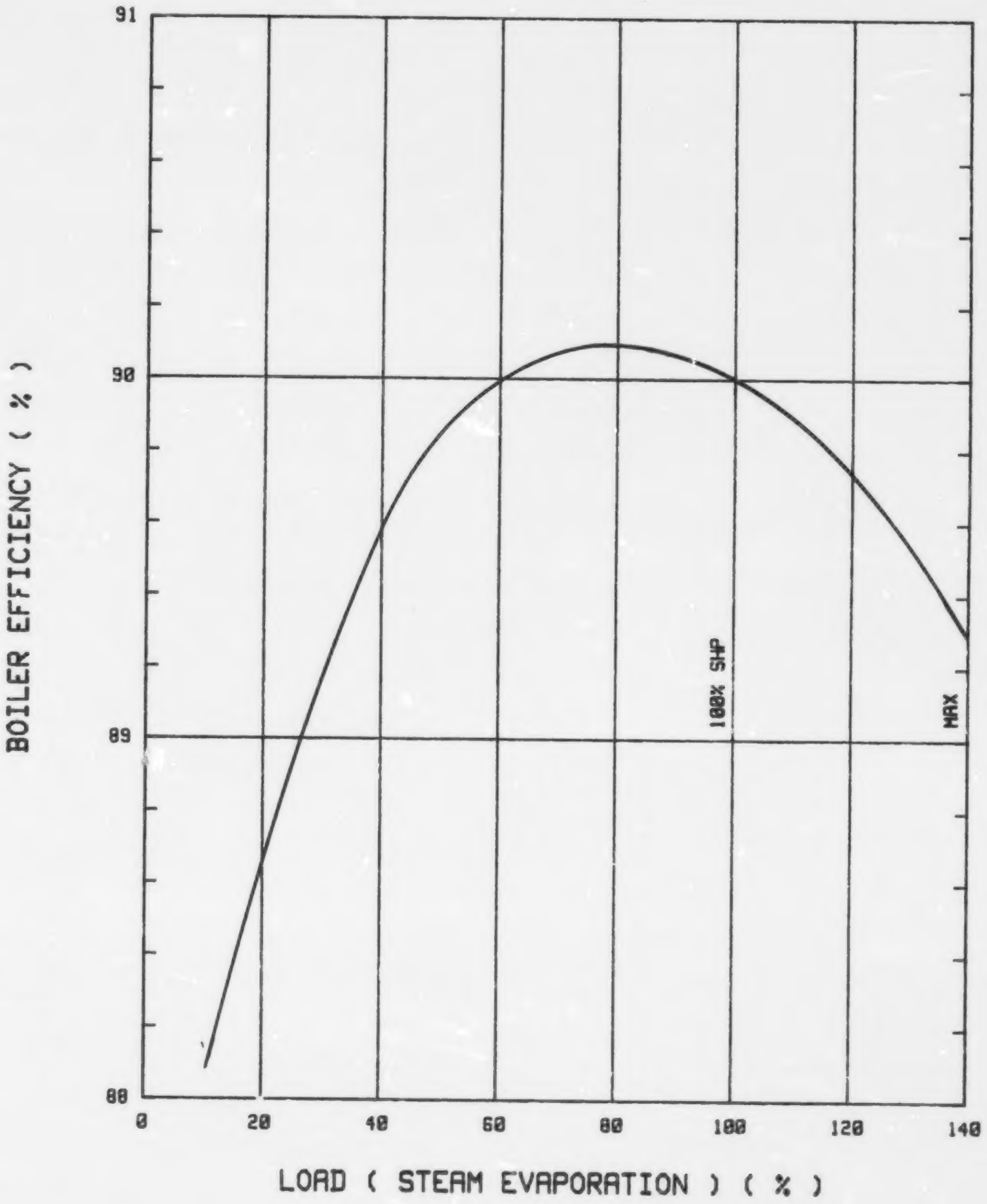
SPECIFIC FUEL CONSUMPTION OF STEAM TURBINE PLANT



[33]

FIGURE 3.10

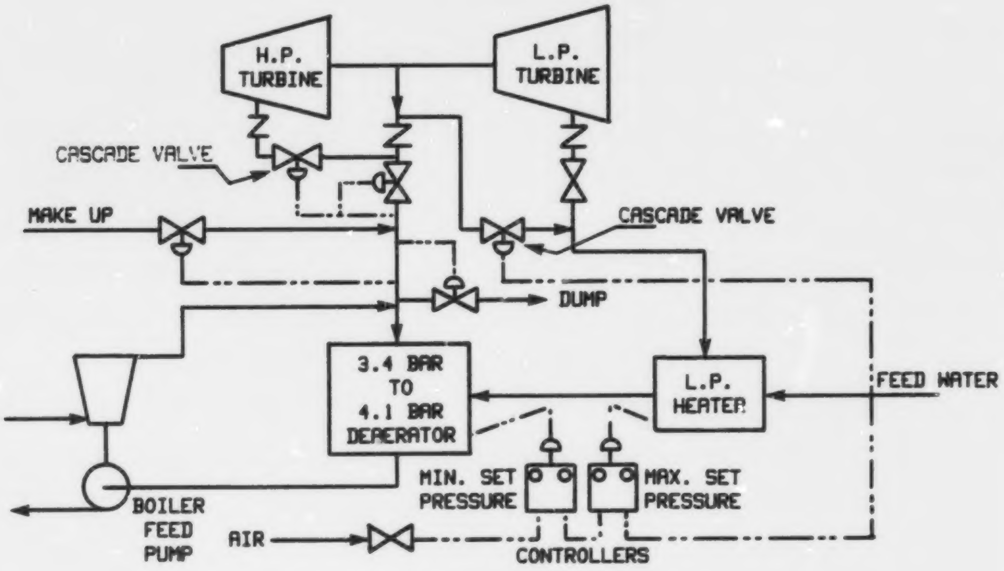
BOILER EFFICIENCY AS A FUNCTION OF LOAD



[33]

FIGURE 3.11

SCHEMATIC DIAGRAM OF CASCADE EXTRACTION SYSTEM


H.P. TURBINE
L.P. TURBINE
CASCADE VALVE
MAKE UP
CASCADE VALVE
DUMP
3.4 BAR TO 4.1 BAR DEAERATOR
L.P. HEATER
FEED WATER
BOILER FEED PUMP
MIN. SET PRESSURE
MAX. SET PRESSURE
AIR
CONTROLLERS

[31, 32]

to be determined the flue gases are analysed. The most commonly used monitoring system usually measures the carbon monoxide (CO), unburned hydrocarbons (HC) in the flue gases and opacity (smoke density). CO is the system's prime control parameter in the majority of marine application. As demonstrated in Figure 3.12 a curve of CO versus relative air flow is sharply "knuckled" in the region of optimum air/fuel ratio. This inherent characteristic enables the small trace of CO generated in an efficient fire to be a sensitive control parameter for combustion air supply regulation. A rise in HC above normal is usually an indicator of improper fuel atomization. Changes in opacity generally occur during large load swings and apart from fuel loss, can lead to air pollution citations in port if not prevented.

Recently developed flue-gas analyzers detect CO and HC by a spectrographic analysis technique using an infrared light source. Opacity is detected with a visual light. The light beams are projected across the centre of the flue uptake to a retroreflecting mirror and are then returned to the flue gas analyzer, which converts the light signal into an electronic data display and control output. The intensity of signals received is calibrated to indicate in parts per million for CO and HC, and percent for opacity. An example of such an analyser is the OTOX system developed by Neotronics Ltd., Takeley, U.K.

The signals from the fuel gas analysis are fed to the combustion control which automatically regulates forced draft fan air delivery to maintain peak combustion efficiency under all variables of fuel characteristics, atmospheric changes and boiler loads.

Some American containerships demonstrated large fuel oil savings by applying this control system and though the initial cost of sophisticated equipment is comparatively high, it can be recovered within several months of operation [95].

To achieve the optimum fuel efficiency, it is necessary to carefully control and monitor not only boiler but also the turbine and auxiliary systems performance. Figure 3.13 shows the parameters which are automatically measured and then evaluated in the control microcomputers in the Stal-Laval prototype VAP Plant. Continuous monitoring of pressures and temperature often reveal various losses within a plant. Readings and trend graphs compared with the original trial conditions allows a true evaluation and improvement in overall economy [96]. A commercially available system, Marine Boiler Automatic Coombustion Control System (MACCS)-1600, has been introduced recently by Mitsubishi Heavy Industries, Ltd., Tokyo, Japan.

3.4 Summary

At the present time propulsion power systems of merchant ships all over the world are almost entirely dominated by three types of machinery: direct drive slow speed diesel engines, geared medium speed diesel engines; and geared steam turbines.

Before the oil crisis much emphasis was placed on developing these types of machinery, with respect to ease of maintenance and to suit automatic control and unattended operation. The escalation of fuel prices left engine manufacturers facing a market demand for engines with high fuel efficiency. Over the last few years remarkable improvements in engine efficiency have taken place. Slow speed diesel engines with specific fuel consumption, as low as 180g/kwh, are now available, and even further reductions are claimed for the new designs being developed.

Medium speed diesel engines with advantages such as; reduced weight and space requirments; as well as the freedom to choose the optimum propeller r.p.m. through a reduction gear; and the recently developed capability of burning heavy and inexpensive fuels, have become a real alternative to the slow speed diesels despite the fact that their specific fuel consumption is about 10% higher.

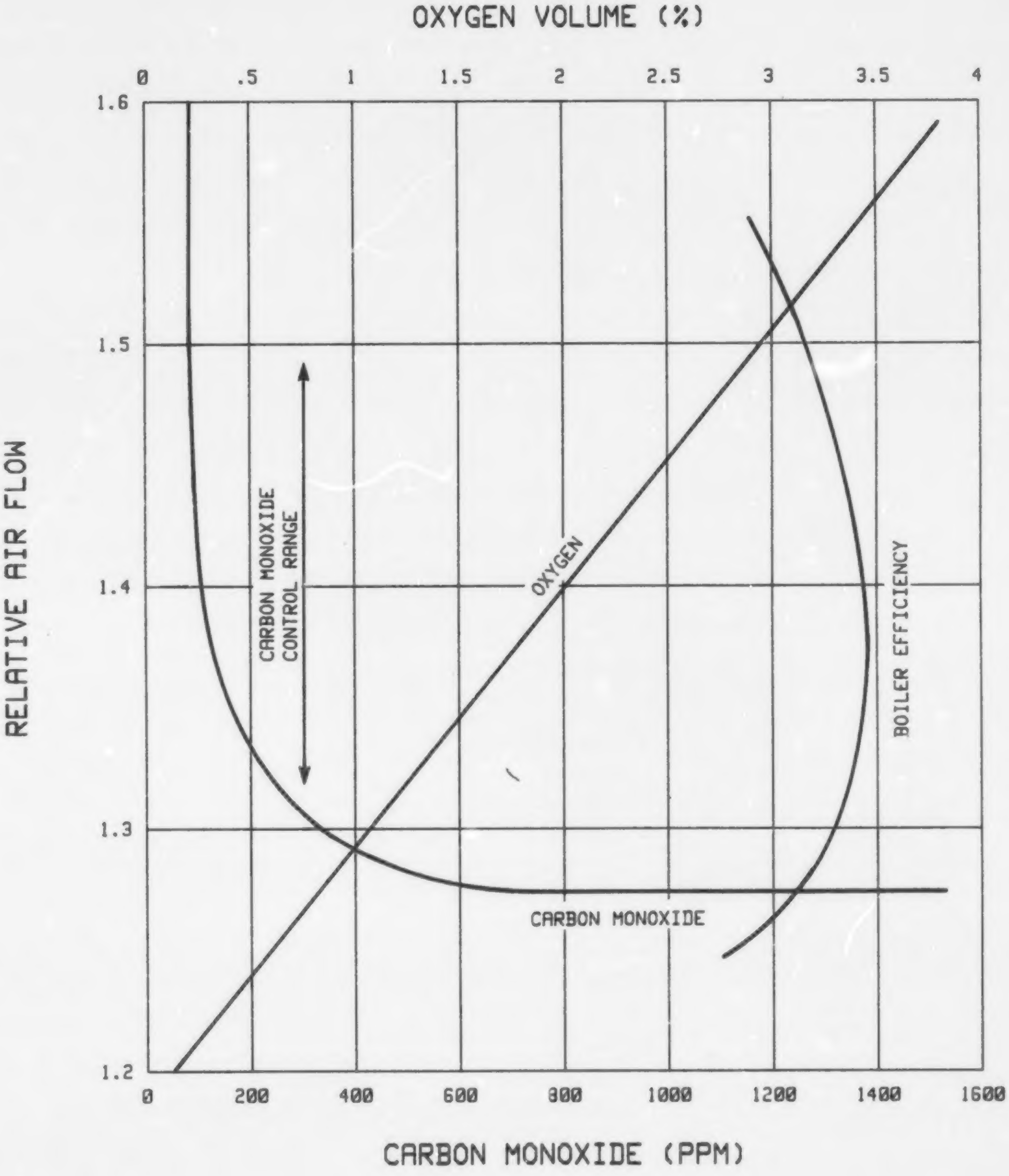
Reductions in fuel consumption were achieved through fresh optimism on the combustion and scavenging processes, together with the use of high efficiency turbo chargers. The most important features of these new solutions are constant pressure turbo-charging and the uniflow scavenging system, which were already well known, but recent computer analytical methods proved their superiority over other systems.

Higher injection pressures up to 1,200 bar which gives fuel droplets of 20 micron diameter; new designs of exhaust valves and their control; as well as fuel valves and many other, sometimes small, details complete a picture of the modern diesel engine. Such engines are much better suited for low load operation and may be a substantial source of fuel savings on certain routes where slow steaming is required.

Similarly, steam turbines have undergone a substantial change in their design and fuel consumption. However, their efficiency is still lower than diesel engines especially at part loads.

FIGURE 3.12

CARBON MONOXIDE CONTENT IN FLUE GAS VERSUS RELATIVE AIR FLOW



[95]

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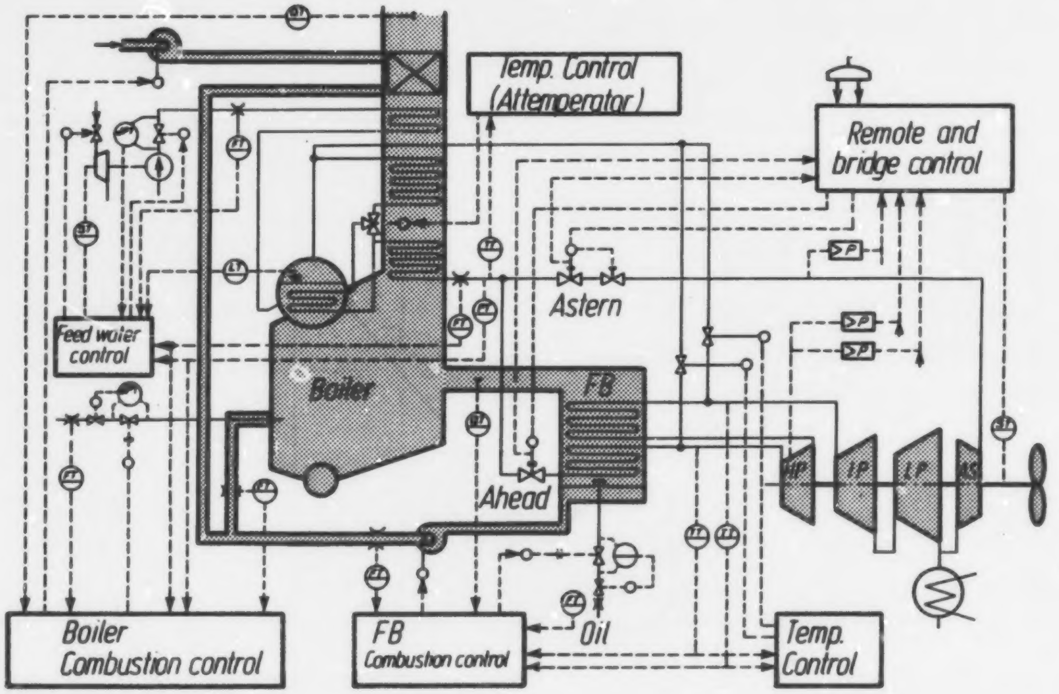
FIGURE 3.13

MAIN SYSTEM PROCESS CONTROL IN STAL-LAVAL V.A.P. PLANT


Temp. Control (Attemperator)
Remote and bridge control
Feed water control
Boiler
FB
Astern
Ahead
ΔP
ΔP
ΔP
HP
IP
LP
AS
Oil
Boiler Combustion control
FB Combustion control
Temp. Control

[96]

4.0 THE THRUST SYSTEM

4.1 General

The efficiency of the ship as a system depends not only on the efficiency of the internal parts (engine and power transmission), but to an equal extent on the external parts; the thrust converting system. This section discusses the thrust system with a view to identifying relationships and equipment effecting fuel efficiency.

4.2 The Conventional Propeller

To improve the propeller efficiency, it is pertinent to review the basis and practice currently adopted for propeller design. The propeller operates by imparting acceleration to the mass of water passing through it to produce thrust. This thrust can be obtained by either:

- Imparting a large acceleration to a small mass of water
- Imparting a small acceleration to a large mass of water

The process of imparting a large acceleration to a small mass of water will require a large force over the swept area, which would involve either a high blade area or high blade loading (pressure), both of which would lead to a drop in efficiency due to wake energy loss and cavitation loss. The development of thrust can be more efficiently achieved by the small acceleration of a large mass of water.

Propeller efficiency is influenced by the following parameters:

- Propeller diameter: the bigger the better with a maximum imposed by stern frame aperture and draft restrictions
- Propeller revolutions: the lower the better with a limitation on the maximum diameter of propeller allowable
- Number of blades: the fewer the better with the choice limited by vibration problems
- Blade area: the smaller the better with the minimum imposed by cavitation problems
- Blade surface roughness: the smoother the better

4.2.1 Large Diameter and Low Reving Propellers

Improvement of efficiency by use of a large diameter and slow speed propeller is based on deriving thrust by applying a small acceleration to a large mass of water. Following this principle the propeller diameter should be as large as practically possible. As the diameter increases, the necessary rotational speed required to maintain the same thrust will reduce.

Figures 4.1 and 4.2 show typical relationships between the propeller diameter, efficiency and r.p.m. A typical diagram commonly used for propeller design is shown in Figure 4.3. From this, it is obvious that any decrease in the r.p.m. for a given P/D will result in a shift to the left resulting in a higher efficiency value.

In pursuing the principle of adopting the largest possible diameter and corresponding lower revolutions, the effect this will have upon the other QPC components must also be considered. With increasing diameter, the effective wake will reduce and the thrust deduction factor will increase due to closer proximity of the propeller to the hull. As a result of this, the hull efficiency components will reduce with a consequent reduction in the QPC improvement that would be obtained from the increase in propeller efficiency alone.

Results from tests of a 300,000 dwt tanker having its original six bladed 9.2 m diameter replaced by a four bladed propeller of 10.5 m diameter showed that although a substantial open water efficiency improvement was achieved, only a 10% QPC improvement was obtained due to a lower hull efficiency [35].

This large diameter approach is not a new principle, the British propeller manufacturer, Stone Manganese Marine, recommended it 20 years ago, but it is only more recently that is has become accepted as a solution. Although it has some problems, such as hull interaction and extra weight, this option is practical and offers significant benefits.

4.2.2 Number of Propeller Blades

Normally, the number of blades is not chosen on the basis of maximum efficiency, but mainly from consideration of vibrations. Whilst recognizing this, it is an important fact that for a given r.p.m., the lower the number of blades the greater the propeller efficiency. Given in Figure 4.4 is a typical relationship between efficiency and number of propeller blades. This figure was reproduced for illustrative purposes for a Wageningen B-Series propeller.

In situations where lower r.p.m. has to be traded off for lower blade number, a higher blade number can be the more efficient alternative. This is because the optimum revolutions are lower for a higher number of blades for a given diameter restriction and in such cases advantage can be taken of the associated increase in efficiency.

4.2.3 Blade Area

The blade surface area of a propeller, and its distribution, has a direct effect upon its efficiency. This is associated with the surface friction caused by the propeller in action. The choice of blade surface area is, however, limited by the requirement to achieve an acceptable cavitation performance under all operating conditions. A clear understanding is required of the relationship between surface area and cavitation properties, aiming at a minimum blade surface area within the cavitation requirement.

A typical relationship between efficiency and blade area is shown in Figure 4.5. The efficiency gain over a blade area ratio range from 0.6 to 1.0 is approximately 2%.

FIGURE 4.1

TYPICAL PROPELLER DIAMETER FOR VARYING R.P.M.

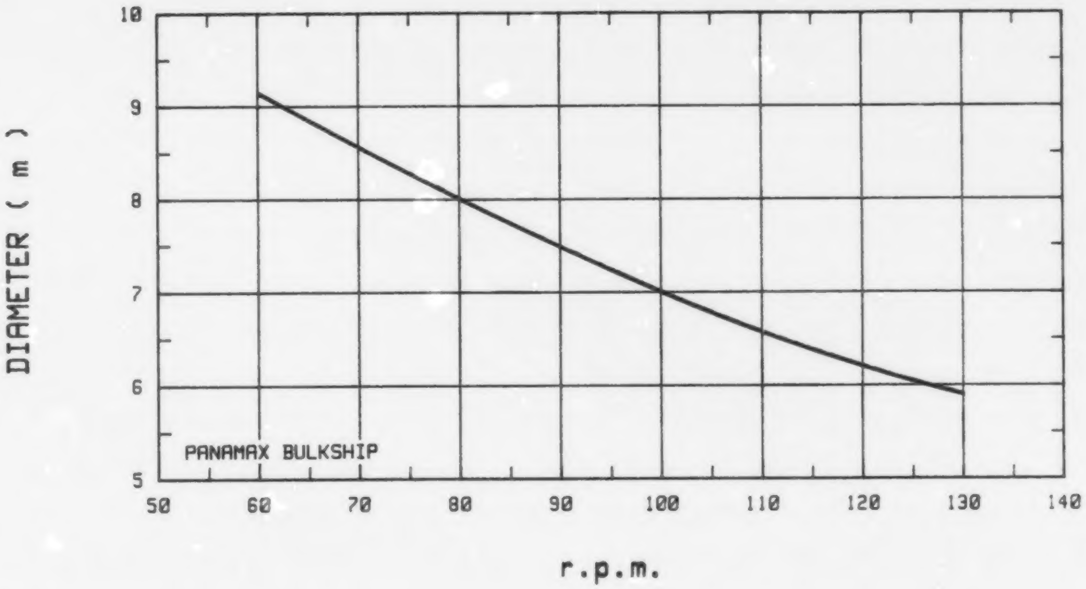


FIGURE 4.2

TYPICAL PROPELLER EFFICIENCY FOR VARYING R.P.M.

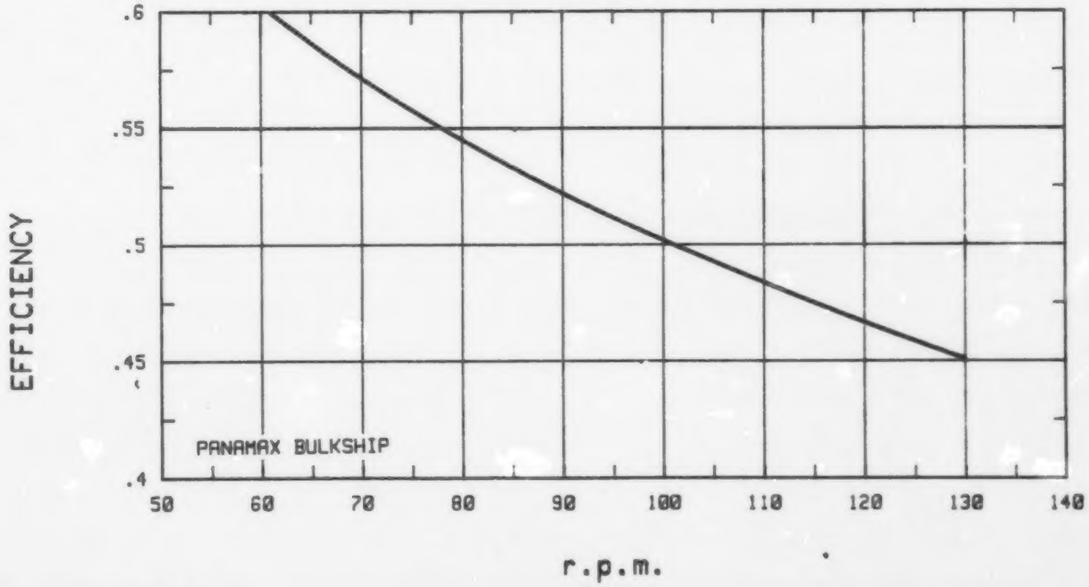
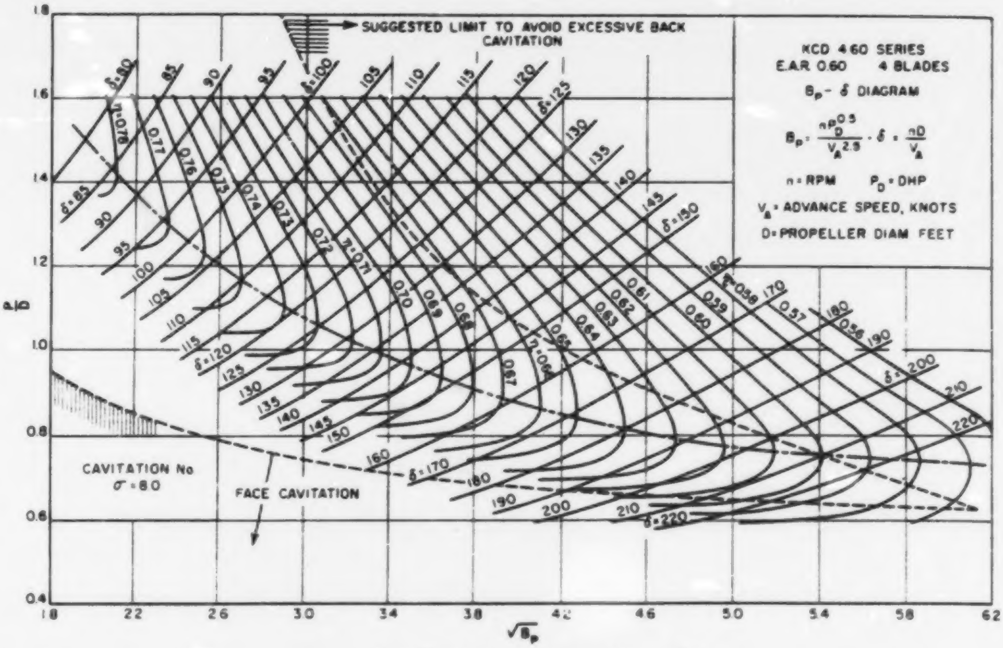


FIGURE 4.3

TYPICAL DIAGRAM USED FOR PROPELLER DESIGN



$B_P - \delta$ diagram for cavitation number $\sigma = 8.0$

FIGURE 4.4

BLADE NUMBER VS. EFFICIENCY FOR A B-SERIES PROPELLER

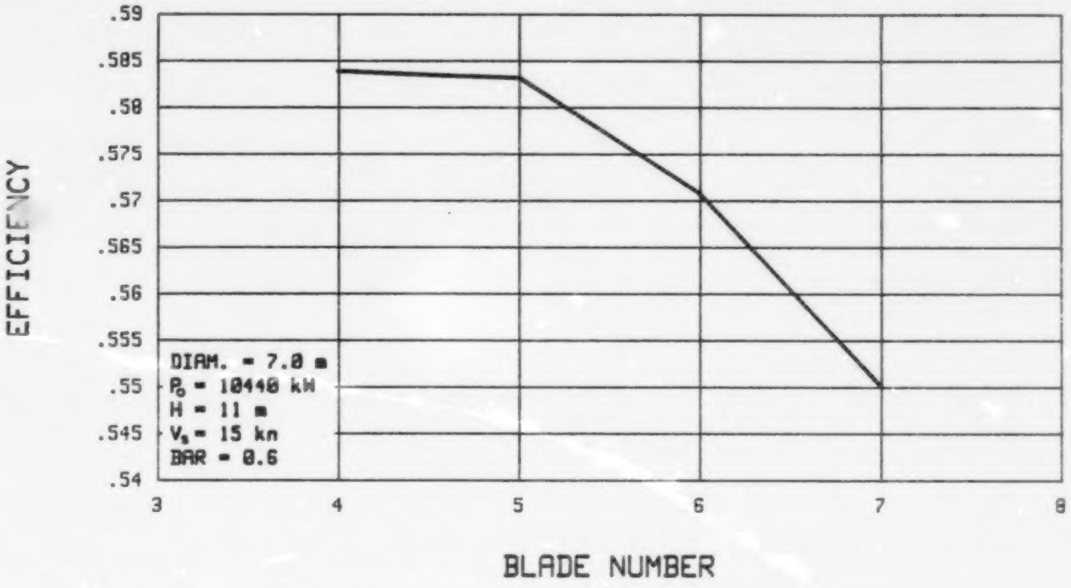
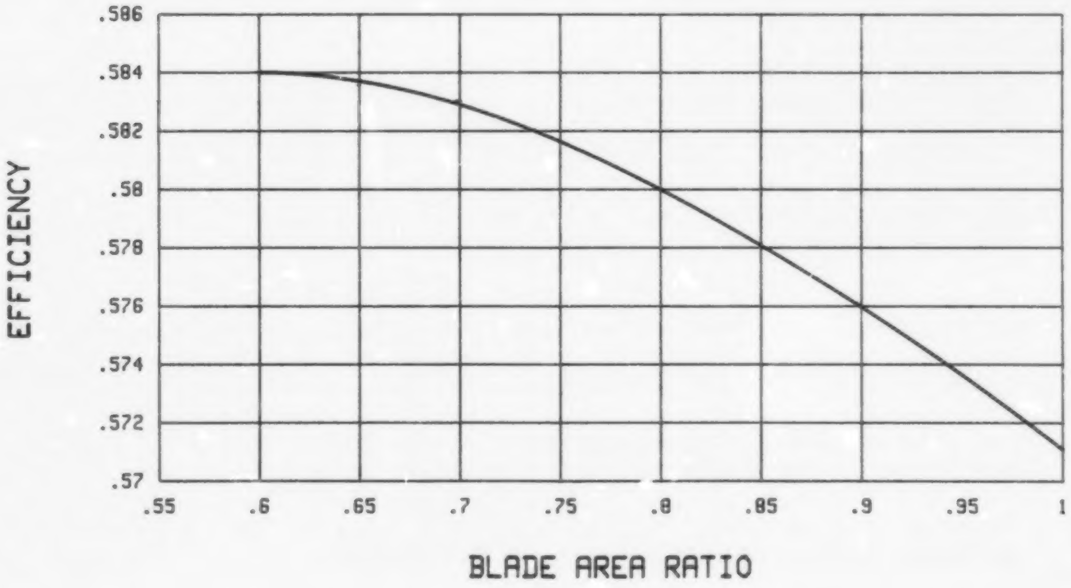


FIGURE 4.5

BLADE AREA RATIO VS. EFFICIENCY FOR A B-SERIES PROPELLER



4.2.4 Propeller Roughness

The surface finish of propeller blades is an important factor in propulsive efficiency, for both new and old propellers. Although the propeller has a relatively small area, the energy losses per unit area, due to blade roughness, are potentially large. At the same time, while it is intuitively accepted that propeller roughness will have an adverse effect on ship performance, little organized information is available for effectively tackling this problem. The lack of information is mainly due to the confusion in defining roughness itself.

Figure 4.6 shows the percentage loss in power due to roughness in the propeller of a 30,000 dwt container ship at two ship conditions. It can be observed that an energy saving of up to 5.5% can be made.

The sensitivity of power loss to where the roughness exists should also be considered. To demonstrate this point, Figure 4.7 is presented. The greatest saving can be obtained by polishing the outer half of the blade. Most well maintained propellers have ratios of R_{tm} (2.5)/$_t$ of 25 or less, at lower values of this ratio the percentage power saving by polishing the outer half of the blade, as opposed to the whole blade is usually better than 83%.

The propeller blades should be polished as often as opportunity allows and special attention should be given to the outer blade region. If possible, the blade tips could be polished afloat with the vessel suitably trimmed to allow access, since drydocking periods of two years and up to four years are now accepted practice.

4.3 TVF (Tip Vortex Free) Propellers

Modern theoretical methods of propeller design are based on the vortex theory first enunciated by F.W. Lancaster in 1907 in his treatise "Aerial Flight" which describes how "lift" is generated due to the presence of circular flow.

The lift produced by a propeller blade is the result of an increased pressure on the face and decreased pressure on the back. The fluid flow follows the pressure gradient and tends to spill over the free ends from the face to the back creating powerful vortices downstream with its axis practically at right angles to the axis of the blade. It follows then that the lift of the blade decreases to zero at the ends due to the tip vortices.

It is clear that total thrust can be increased if energy previously wasted on frictional loss and vortex loss can be utilized. Recent progress has been made by the introduction of TVF propellers, most notably by Astilleros Espanoles S.A. of Spain (A.E.S.A.).

The general method adopted to avoid the formation of tip vortices at the ends of propeller blades has been the modification of the geometry of blade tips by either adding a tip blade as in the case of the A.E.S.A. design or a rotating ring connecting all the blade tips as with the "Church Propellers" developed by Propeller Design Ltd. of New Zealand.

FIGURE 4.6

PERCENTAGE LOSS IN POWER DUE TO ROUGHENED PROPELLER

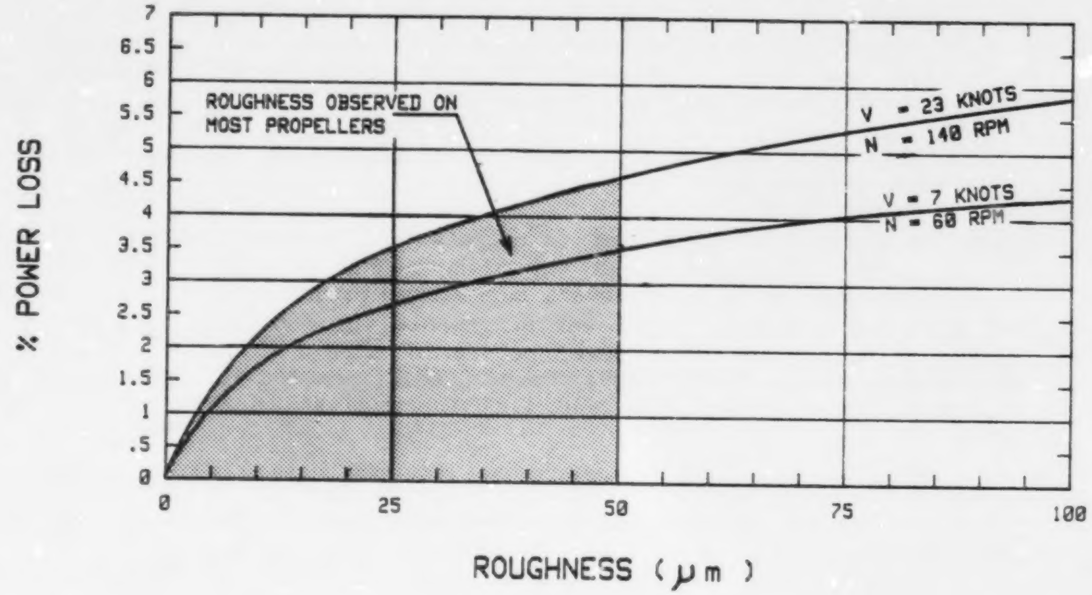
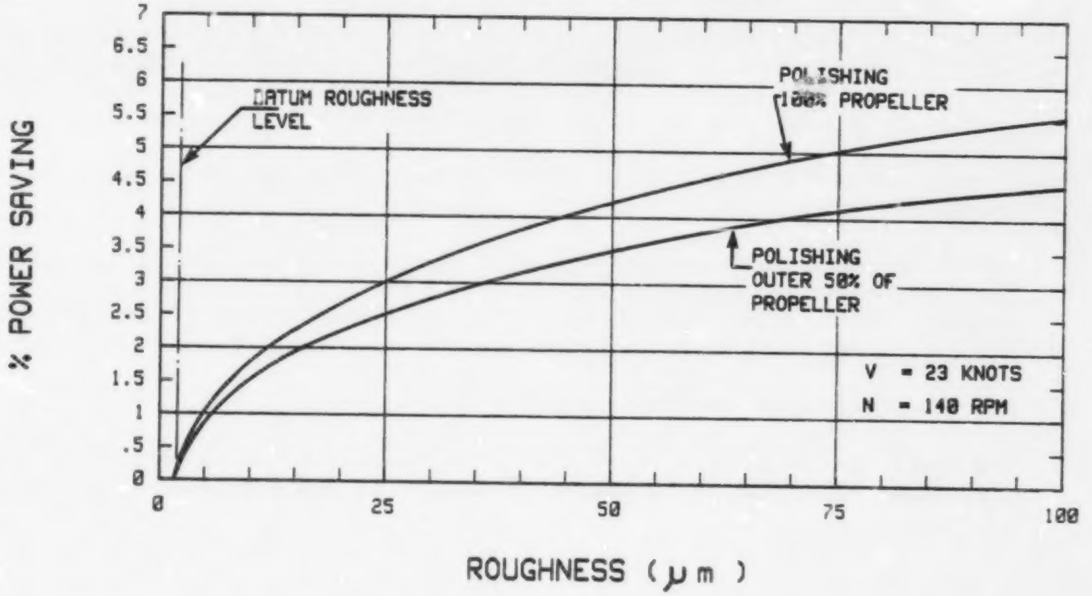


FIGURE 4.7

PERCENTAGE SAVING IN POWER DUE TO POLISHING THE PROPELLER TO A DATUM ROUGHNESS LEVEL



Based on the findings of A.E.S.A., the advantages for TVF propellers are:

- an efficiency in open water markedly higher than conventional propellers
- an optimum diameter less than that of conventional propellers and therefore, despite their greater peripheral blade area, their weight and also their axial inertia moments are of the same magnitude making them particularly advantageous as replacements for existing conventional propellers
- an open water efficiency better than conventional propellers in a nozzle while they do not cause cavitation erosion of the inside of the propeller duct

In spite of the theoretical merits and encouraging model and full scale test results, the TVF propeller is not without problems, as indicated by full scale test results [36]. However, overall performance results showed that a power saving of 24% to 14.3% was achievable for "SOKORRI", the 35,000 dwt bulk carrier tested and a power saving of around 15% in the case of the 270,000 dwt tanker "RIO TINTO". Power predictions and full scale results of tests on the "SOKORRI" and "RIO TINTO" are given in Figures 4.8 and 4.9 [36].

To prevent flow separation at the closing plates on the blade tips, propeller ducts have been used in conjunction with the TVF propellers in all open water tests. Extensive hull modification, to improve the flow configuration, was found necessary to achieve satisfactory on-ship performance.

4.4 Contra-Rotating Propellers

The principle of contra-rotating propellers has been the subject of some considerable research in the past and is of great advantage for fast ships with restricted draft (and therefore propeller diameter). Under such situations where the rotational wake energy imparted by the propeller is generally high, the utilization of a rear screw for this rotational energy recovery becomes significant. Typically, earlier investigations indicate that a power reduction of 10% can be achieved over a single propeller [37].

A more recent application of this concept gave an improvement in fuel economy of 10% to 15%.

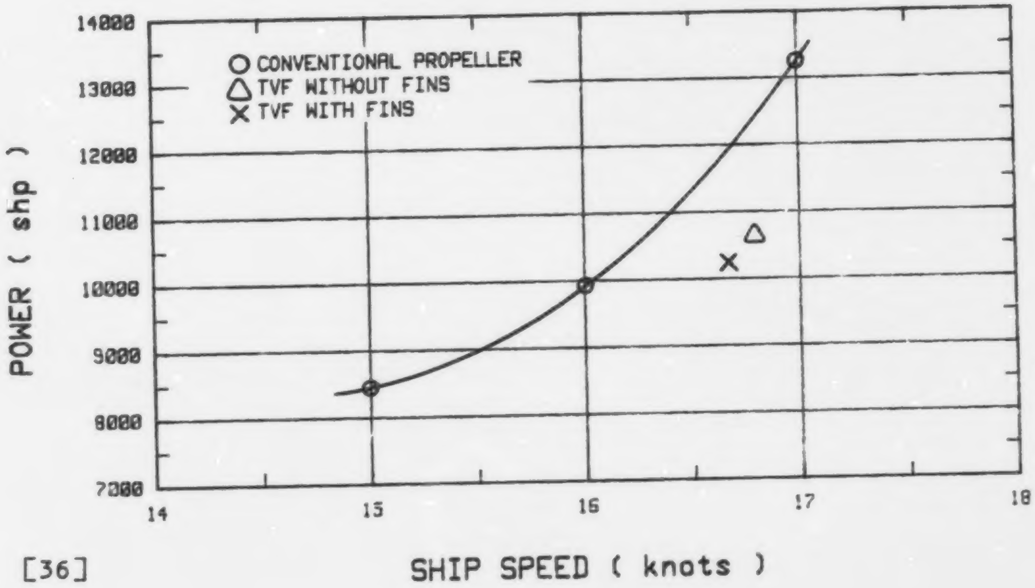
For large ship applications, the main reason why this concept has not been put into common application, is the mechanical complexity involved. However, with ever increasing fuel cost, the case of contra-rotating propellers may progressively become a worthwhile case of consideration.

4.5 Reaction Fins/Reaction Propellers

The reaction fin concept is to provide a prewhirl to the propeller inflow to give a net reduction in rotational flow left by the propeller slip stream.

FIGURE 4.8

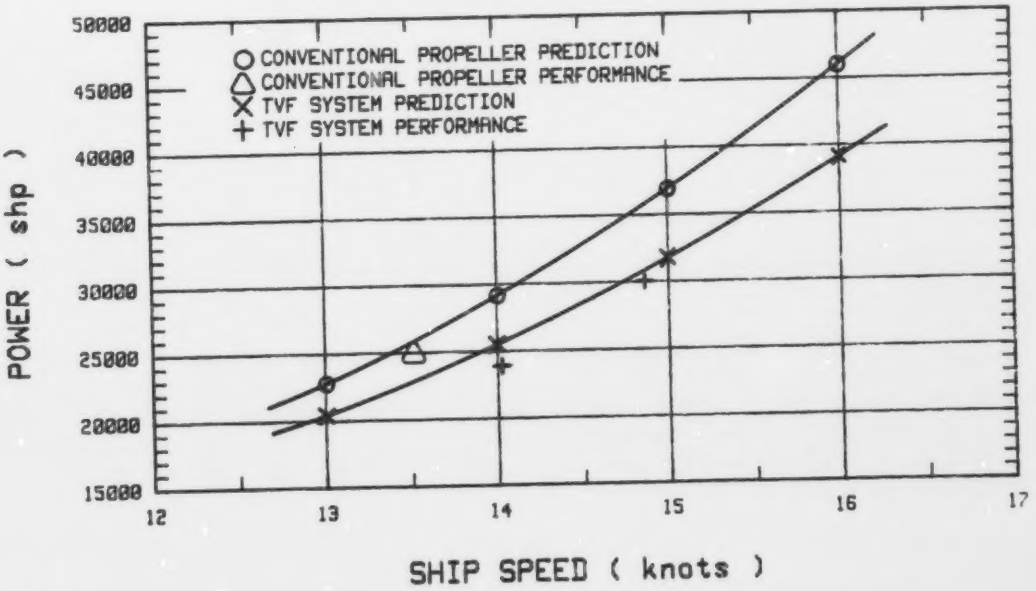
SEA TRIALS RESULTS OF 'SOKORRI'



[36]

FIGURE 4.9

POWER PREDICTIONS AND FULL SCALE
RESULTS OF 'RIO-TINTO'



[36]

This type of device has been fitted on a number of vessels, one up to 270,000 dwt, with power savings of between 4 to 8%. The configuration is shown in Figure 4.10.

A variation of the reaction fin is the reaction propeller, as illustrated in Figure 4.11. For the reaction propeller, it is claimed that around five tons of fuel per day should be cut from the consumption of a 75,000 dwt ship. In this application, a 16 tonne, 7.5 m diameter, nine bladed reaction propeller is fitted to redistribute the energy absorbed from the central field of the screw wake to the outer diameter of the main propeller [38].

4.6 Ducted Propellers

Propellers operating in nozzles/ducts show significant advantages when fitted to tugs, trawlers and icebreakers where high thrust is required at low speeds. The same principle can be applied to ships of large mass, e.g., tankers and bulk carriers, that operate at relatively low speeds and where the thrust requirement is high. With ducted propellers, gains in thrust over the open propeller of between 30% and 45% are possible at low speeds, due to the positive contribution from the duct at low advance speeds. As speed increases, this contribution diminishes and at higher speeds can even be negative.

Therefore, under certain circumstances, the propeller/nozzle system can give rise to an improvement in the overall propulsive efficiency.

Generally, the degree of loading, which is measured by C_T, the specific propeller loading coefficient, can be used as the parameter for assessing on the suitability of particular duct applications. Typical C_T values for different ship types are given as follows [40]:

Ship Type	C_T
Twin Screw Ships	0.5 - 1.0
Cargo Ships	1.0 - 3.0
Tankers	2.5 - 5.0
Trawlers	4.0 - 8.0
Tugs	> 6.0

It can be seen that for ship types with higher C_T values, ducts are quite common.

For a particular propeller diameter and revolution, the open water efficiency of a propeller follows a trend of decreasing efficiency with decreasing design speed over a certain speed range. The ducted propeller also exhibits this characteristic. However, for low design speeds the ducted propeller system generally has a higher efficiency compared with an equivalent open propeller.

FIGURE 4.10

SCHEMATIC VIEW OF A REACTION FIN SYSTEM

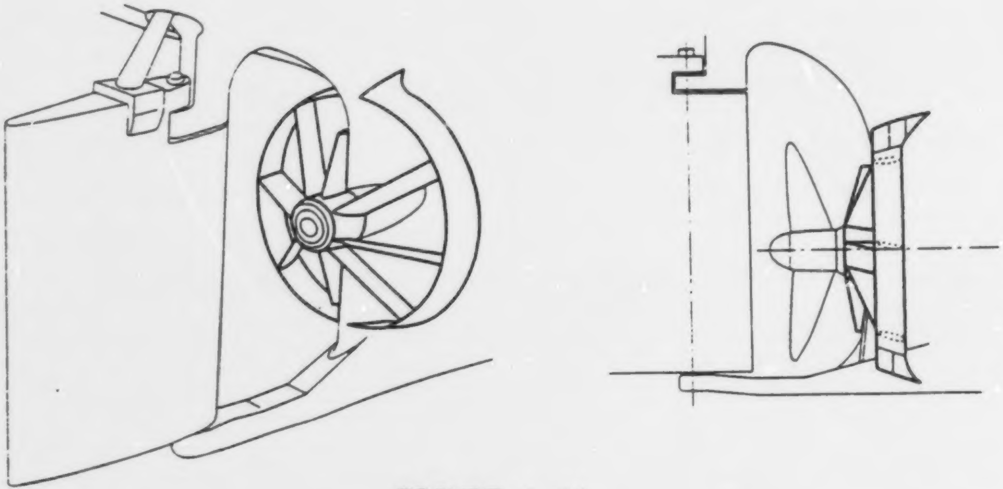
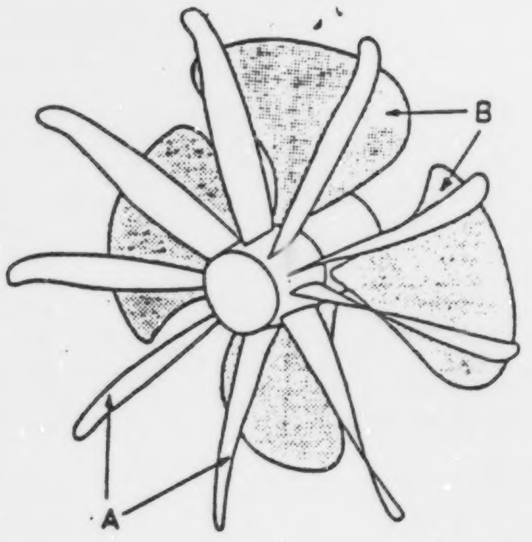


FIGURE 4.11

[38] SCHEMATIC VIEW OF A REACTION PROPELLER



Around five tons of fuel a day should be cut from the consumption of a 75,000 dwt multipurpose freighter completing at the Bremer Vulkan shipyard in West Germany. Directly behind the propeller (B in diagram) is fitted a nine bladed reaction propeller (A). The 16 ton, 7.5 m diameter reaction propeller is the invention of Professor Otto Grim of Hamburg. It leads energy absorbed from the central field of the screw wake to the outer diameter of the main propeller.

[38]

Figure 4.12 shows the comparison of an open Wageningen B4-70 propeller and a system with a B4-70 in a 19A duct. The open propeller is assumed to be of 6 m diameter at a constant design speed of 120 r.p.m., while the ducted propeller has a smaller diameter of 5.2 m to account for the space taken up by the duct in a fixed stern aperture situation. The design speed of the ducted propeller is 143 r.p.m. Data for this comparison is obtained from references [152, 153]. It can be observed from this figure that at low speeds of up to approximately 12 knots, the ducted system gives a higher efficiency than the open propeller. Beyond 12 knots, the drag of the duct outweighs the efficiency gain and returns a negative net gain. What this particular comparison indicates is that the 5.2 m diameter B4-70 ducted propeller system is only suitable for application for ships with design speeds of less than 12 knots. This breakeven speed can be varied by choosing a nozzle with different geometric properties. For higher speed applications, a thinner nozzle may be used. A thinner nozzle will have less drag at higher speeds, but at the same time will not provide the same increase of thrust at low speeds. From Figure 4.12, Figure 4.13 can be produced to indicate the highest efficiency gain is at a design speed of approximately 4 knots. However, in the actual design process, other constraints may have to be taken into account.

Table 4.1 gives the fuel saving on a number of ships through the use of a suitable duct. The fuel saving is in the range of 5% to 8% for the selected examples [39].

An economic analysis of the use of a ducted propeller system on a 32,400 tonne displacement bulk carrier performed in 1977 shows that the additional capital cost involved in fitting a ducted propeller can be recovered in approximately 1.5 years through a power reduction of 5.44%, showing that the ducted propeller system can be an energy efficient and economically attractive investment in the correct application [41].

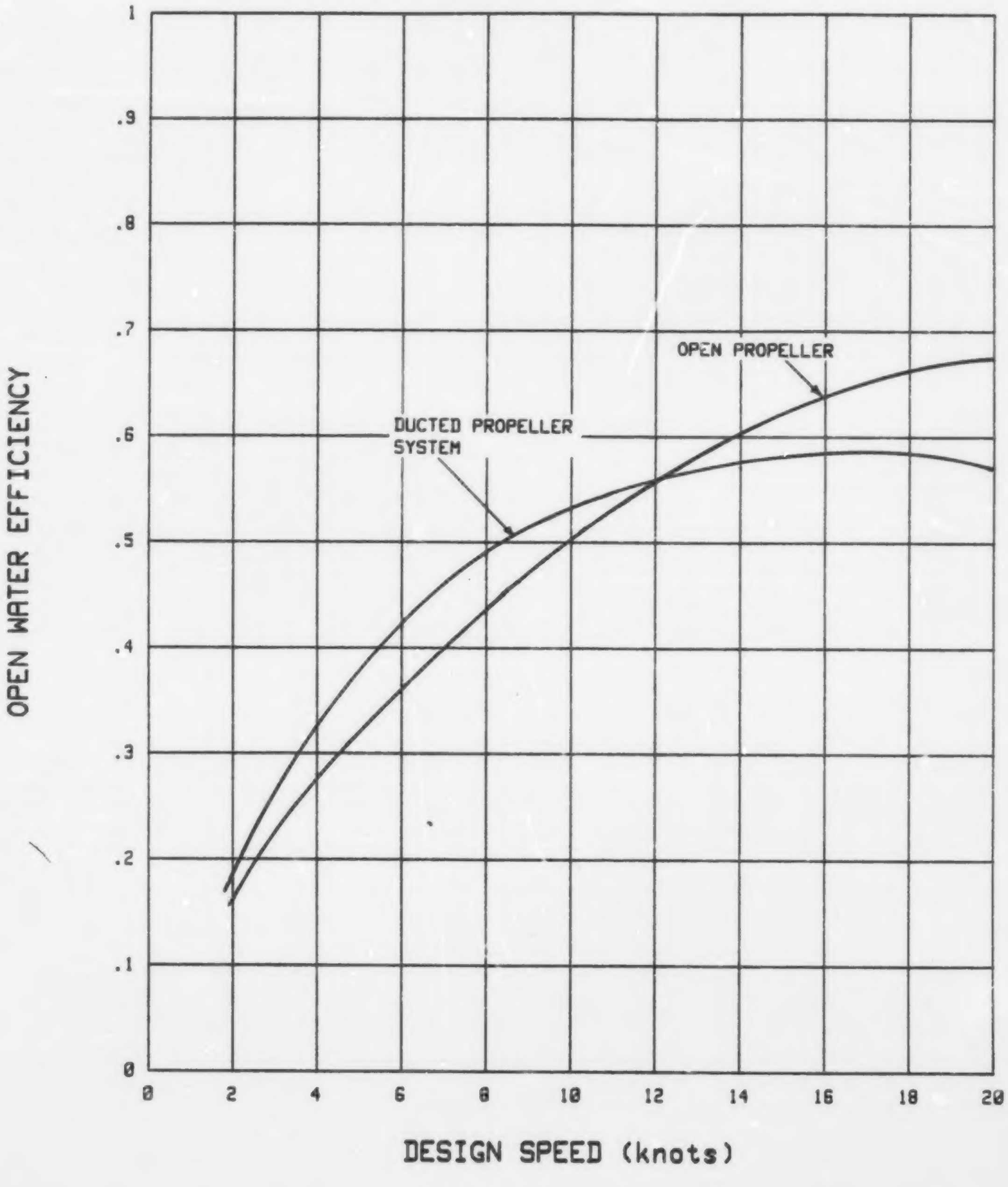
4.7 Controllable Pitch Propellers

The controllable pitch propeller has been widely used in recent years. In the case of large merchant ships such as tankers, CP propellers are fitted mainly for advantages in flexibility and control.

CP propellers require a larger boss resulting in a restriction of the design of the blade root and unfavourable angles. This causes the CP propeller to be less efficient than a fixed pitch propeller at the design condition but at off-design conditions CP propellers may be of equal or greater efficiency than the fixed pitch system. However, in spite of the inherent lower efficiency, CP propellers have found application in a wide range of ships for various reasons. On trawlers, tugs, etc. they provide an increase in bollard pull of about 30 - 40% without greatly affecting the free running performance. On ferries and warships, acceleration, deceleration and stopping distance can be halved with CP propellers capable of absorbing full MCR power within 10 to 30 seconds of the "full astern" signal and there is no shaft momentum loss in such manoeuvres.

FIGURE 4.12

COMPARISON OF OPEN WATER EFFICIENCY OF OPEN PROPELLER AND DUCTED PROPELLER SYSTEM

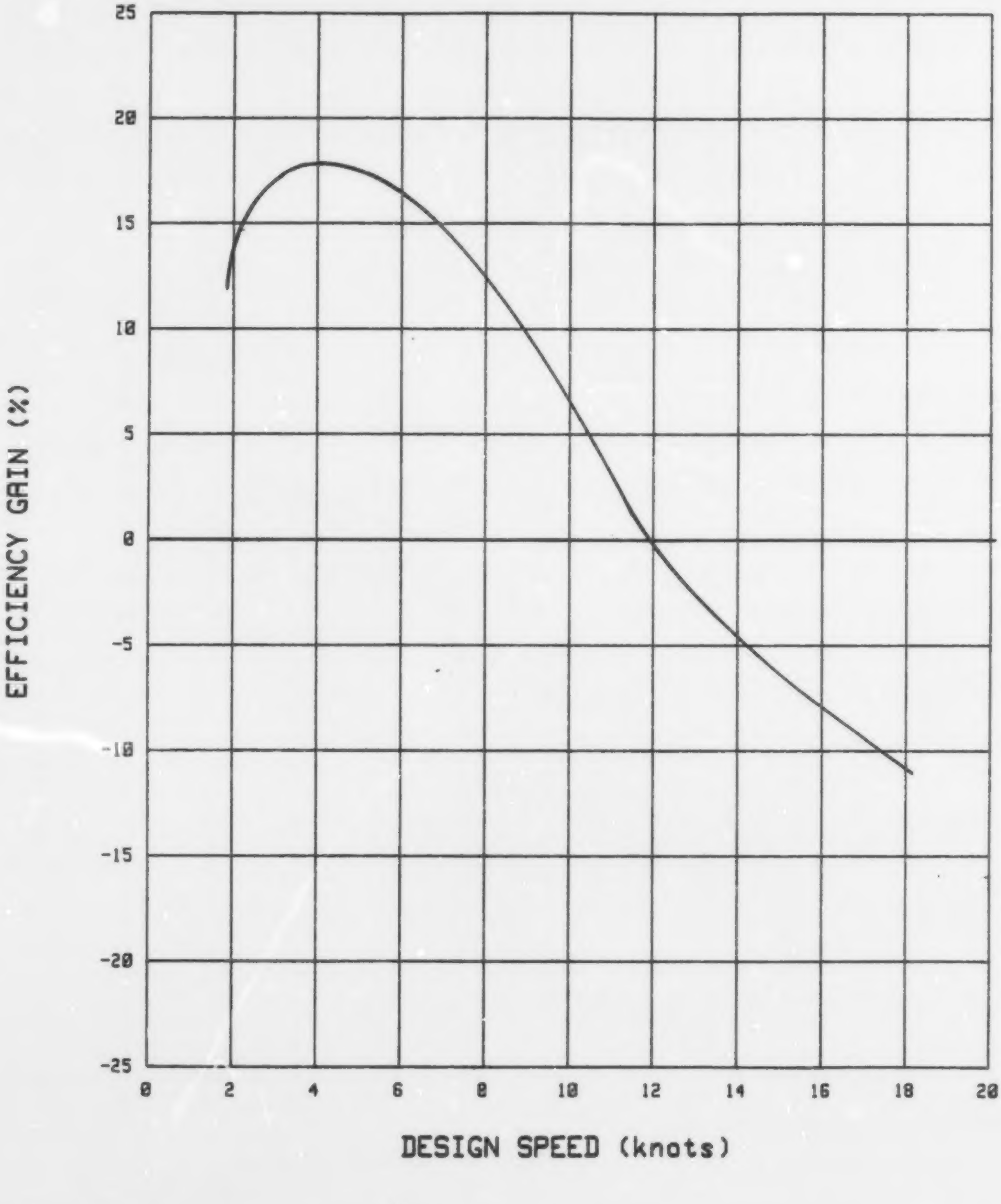


*OPEN PROPELLER = 6m DIAMETER WAGENINGEN B4-70 @ 120 rpm
DUCTED PROPELLER = 5.2 m DIAMETER WAGENINGEN B4-70 IN 19A NOZZLE @ 143 rpm

MELVILLE SHIPPING LTD

FIGURE 4.13

EFFICIENCY GAIN OF DUCTED PROPELLER SYSTEM OVER OPEN PROPELLER AT VARYING DESIGN SPEEDS



*OPEN PROPELLER = 6m DIAMETER WAGENINGEN B4-70 @ 120 rpm
DUCTED PROPELLER = 5.2 m DIAMETER WAGENINGEN B4-70 IN A19 NOZZLE @ 143 rpm

MELVILLE SHIPPING LTD

TABLE 4.1

ESTIMATED FUEL SAVING BY USING DUCTED PROPELLERS

	LARGE TANKER	DUAL OBO CARRIER	LARGE BULK CARRIER	MEDIUM BULK CARRIER
BASIC SHIP INFORMATION				
Length (m)	352	282	256	175
Deadweight (tonnes)	316,000	169,670	101,600	25,400
Type of Engine	Turbine	Diesel	Diesel	Diesel
Maximum Horse Power	36,000	30,400	26,800	9,900
R.P.M.	85	103	122	150
Ship Speed (knots)	15	15.5	15.9	15
CONVENTIONAL PROPELLER				
Diameter (m)	8.65	7.6	6.5	5.0
Number of Blades	6	6	6	4
DUCTED PROPELLER SYSTEMS				
Propeller:				
Diameter (m)	8.25	7.24	5.7	4.6
Number of Blades	6	6	6	4
ESTIMATED SAVINGS WITH DUCTED PROPELLER SYSTEM				
Fuel Saving % (estimated by model testing)	5.5	8.0	7.7	8.0

[39]

The flexibility obtainable from a propulsion system with a CP propeller forms the very basis of the superiority of the CP propeller over the fixed pitch propeller in situations where such flexibility is required.

To illustrate the significance of flexibility in the propulsion system, if a ship runs on a schedule such that reduced speed is acceptable between some ports of call, then a ship fitted with a CP propeller has an advantage over a ship fitted with a fixed pitch propeller. Figure 4.14 shows a typical set of performance curves relating to engine speed, power output, propeller pitch, fuel consumption and ship speed [42]. In this figure, the condition at which design speed is achieved, at 100% MCR power and 100% rated r.p.m., is represented by Point A. If ship speed is reduced to 90% of the design speed using a fixed pitch propeller, this will result in a fuel consumption of 68% of the maximum at Point B. Assuming that the pitch of the propeller can be changed, the same ship speed can be achieved at Point C with 140% of the design pitch. By doing this the r.p.m. of the engine can be reduced by approximately 14% and the fuel consumption can be lowered from 68% to 65% of maximum resulting in a fuel saving of 3%.

Special computer control devices are now in operation on some ferries, sailing on regular routes in Scandinavia, which keep fuel consumption to a minimum by selecting the most efficient combination of propeller pitch, speed and power to get the ship from port to port on specific schedules, regardless of variation in external conditions.

Besides offering flexibility in the propulsion system, the CP propeller also allows flexibility in auxiliary power arrangement to reduce the overall running cost. One application where the CP propeller can assist in greater efficiency is on a ship fitted with shaft driven alternators. If a fixed pitch propeller is used, various methods must be used to maintain the output voltage and frequency, which are generally based on relatively complex equipment. Some of these impose restrictions of shaft speed variations. In the case of the CP driven ship, the propulsion engine shaft speed can be kept constant thereby providing a simple and reliable solution to frequency control of the shaft driven alternator.

4.8 Hull Appendages

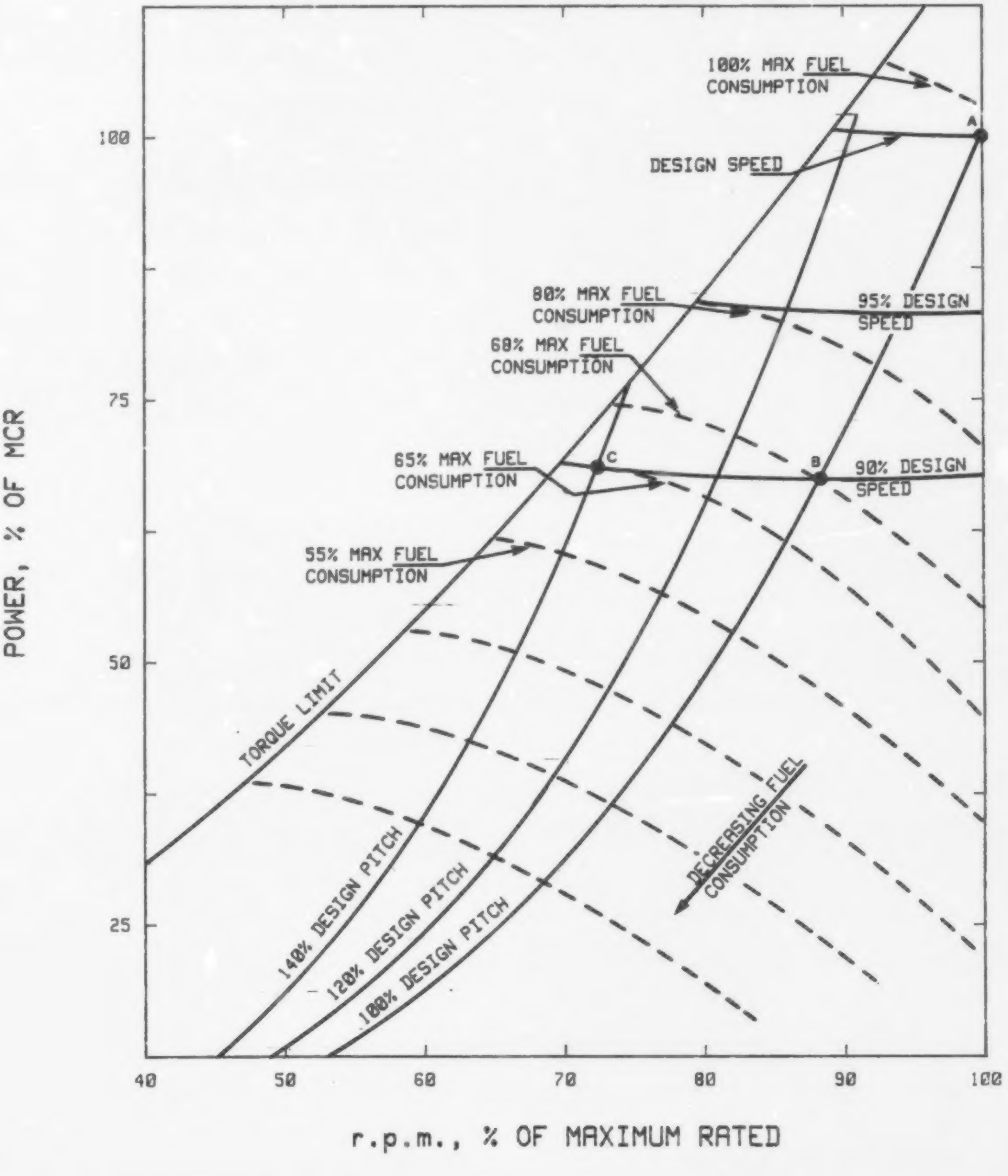
As a contributing factor to the overall Q.P.C. of the external propulsive system, the ship's hull is important. In a drive to achieve higher overall propulsive efficiency, a number of hull appendages have been designed and used in recent years. The general aim is to improve the water flow into the propeller.

These appendages include:

- Flow modifying nozzles, e.g. the Mitsui Integrated Duct Propeller (MIDP), now installed on over 100 ships. See Figure 4.15. The MIDP was installed on the 250,000 dwt tanker Esso Copenhagen and the results were compared with trials in clean hull prior to installation. The power savings attained by this installation are shown in Table 4.2.

FIGURE 4.14

PERFORMANCE CURVES RELATING ENGINE SPEED, POWER OUTPUT, PROPELLER PITCH, FUEL CONSUMPTION AND SHIP SPEED



[42]

FIGURE 4.15

THE MITSUI INTEGRATED DUCT PROPELLER (MIDP)

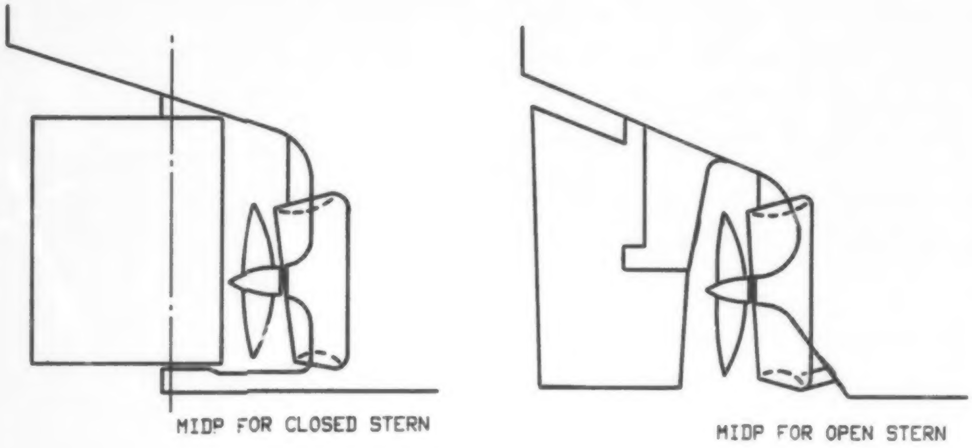

MIDP FOR CLOSED STERN
MIDP FOR OPEN STERN

TABLE 4.2

POWER SAVING BY MIDP

POWER SAVING		
Full Load	at 16 kts	abt. 6%
	at 15 kts	abt. 7%
Ballast	at 18 kts	abt. 4%
	at 14 kts	abt. 10%

SPEED INCREASE		
Full Load	at MCR	0.36 kts
Ballast	at MCR	0.24 kts

- Semi-tunnel sterns, e.g., Port Weller Drydock Ltd. design. The improvement of full-scale test performance is given in Table 4.3.
- Rudder bulb, e.g., Kawasaki design. Figure 4.16 shows a typical outline of the arrangment. The power savings for a 130,000 dwt bulk carrier have been shown to be about 5%.

4.9 Summary

The efficiency of the ship as a system depends not only on the efficiency of the internal parts (engine and power transmission), but to an equal extent on the external parts; the thrust converting system and hull appendages. The conventional fixed pitch open propeller, with aft end lines designed for smooth water flow, are slowly being refined to obtain greater efficiency and modified in some significant ways to overcome recently discovered hydro dynamic effects.

Propeller efficiency is influenced by a) propeller diameter, with larger diameter resulting in higher efficiency, b) propeller revolutions, with lower revolutions resulting in higher efficiency, c) the number of blades, with the fewer number resulting in higher efficiency, d) blade area, with the smaller blade areas resulting in higher efficiency, and e) blade surface roughness, with a smoother surface resulting in signficant efficiency gains. Off these opportunities, propeller r.p.m. (and hence diameter), the number of blades, and propeller roughness, offer the best opportunities for efficiency gains.

In addition to conventional propellers, Tip Vortex Free propellers, Contra-rotating propellers and Reaction Fin propellers, all designed to achieve hydrodynamic benefits, have been adopted by ship owners and can result in significant fuel savings. The real long term benefits of these systems are not yet known nor is their applicability to a wide range of vessels fully understood.

Ducted propellers have been used now for many years, particularly on vessels requiring high bollard pulls, or at least those vessels with generally low operating speeds. The efficiency benefits of these ducted propellers at such low speeds are well recognized as are the draw backs when the operating speed is in the more conventional 14 - 16 knot range. Hull appendages such as the Mitsui Integrated Duct Propeller (MIDP), semi-tunnel sterns and rudder bulbs all offer interesting reductions in fuel consumption. These appendages have been fitted to a variety of vessels, and while the initial results indicate substantial benefits, further experience may be necessary to encourage owners to adopt these systems.

TABLE 4.3

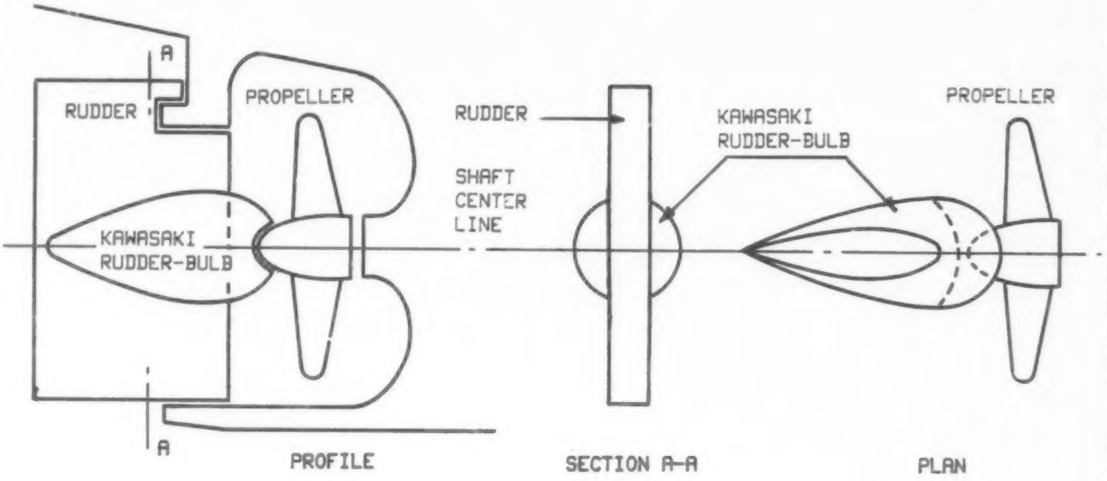
PRINCIPAL PARTICULARS OF PORT WELLER BUILT BULK CARRIERS

VESSEL PARTICULARS	"CANADIAN TRANSPORT" (Conventional)	"CANADIAN ENTERPRISE" (Semi-Tunnel Stern)	HULL 67 (Semi-Tunnel Stern)	HULL 70 (Conventional)
Length, b.p. (m)	220.67	220.67	217.70	217.70
Breadth (m)	23.12	23.12	23.12	23.12
Draught (m)	9.14	9.14	10.50	10.50
Block Coefficient	0.91	0.91	0.89	0.89
Deadweight (tonnes)	28,084*	27,982*	37,500	37,500
Main Engine	2 x MAN 3L40/54A	2 x MAN 7L40/54A	Doxford 76J4	Sulzer 4RLB76
Installed Power (metric bhp)	10,000	8,750	8,000/9,000	8,000/9,000
Gearbox	single reduction	double reduction	none	none
Propeller Diameter (mm)	6,100	7,200	6,650	6,050
Propeller Speed (rev/min)	103	65	85**	99***
Propeller Efficiency	0.51	0.62	0.58	0.57

(*7.92 m Draught) ** 95 rev/min at 9,000 bhp *** 102 rev/min at 9,000 bhp

FIGURE 4.18

SCHEMATIC VIEW OF THE RUDDER BULB SYSTEM


A
RUDDER
PROPELLER
KAWASAKI RUDDER-BULB
A
PROFILE
RUDDER
SHAFT CENTER LINE
SECTION A-A
KAWASAKI RUDDER-BULB
PROPELLER
PLAN

5.0 AUXILIARY SYSTEMS

5.1 General

The design and operation of auxiliary systems contribute in a large part to the energy efficiency of a ship. This section is subdivided into several aspects:

- Auxiliary power systems
- Heating systems
- Waste heat recovery, and
- Pumps

5.2 Auxiliary Power Systems

Auxiliary power requirements for ships can be substantial, depending upon the type of ship. In a bulk carrier fitted with oil engines the auxiliary power required to support the engine systems and basic hotel services is approximately 5% of the installed power. In other types of vessels, in particular passenger liners or cruise ships, the auxiliary power requirements may exceed the propulsion requirements.

The basic requirement to service the pumps and equipment for a steam turbine propulsion system would be approximately 7-9% of the installed power and between 5-7% for a diesel engine propulsion system. In addition to the base electrical load of the propulsion system there may be a considerable additional electrical load due to the bow thruster, if fitted, cargo and ballast pumps, cargo refrigeration, hotel services, air conditioning load, etc. The electrical load required for the basic propulsion system may often be the smallest portion of the total load requirement of the vessel.

Electrical power is normally generated on board ship by the following methods:

- auxiliary diesel engine driven alternator sets
- steam turbine alternator sets
- main engine driven alternator sets

5.2.2 Steam and Diesel Alternators

The large proportion of alternators fitted to ocean going vessels fall within the power range 500-2,000 kW and two or more machines may be fitted to cater for peak load and redundancy requirements. In this power range the efficiency of the oil engine is considerably in excess of the steam turbine, as shown in Table 5.1 and this difference becomes even larger at part load operation.

For diesel engine powered ships the most efficient electric power generating system would be a steam turbine alternator supplied from a waste heat boiler, when at sea, and a heavy oil burning diesel generator for port use. For steam turbine driven ships a heavy oil burning diesel generator is more efficient than a steam turbine driven alternator, although a bled steam driven turbo alternator which exhausts to a direct contact feed heater would be a competitive but more complicated system.

The turbo alternator driven by steam generated by an oil engine exhaust gas boiler offers the most efficient system as no additional fuel is consumed. Of the independent prime mover systems the heavy fuel oil burning diesel engine driven alternators are the cheapest means to generate electrical power although suitable heavy oil burning engines below about 300 kW are not presently available.

At conditions removed from the design point the efficiency gap between turbine driven and the diesel engine driven alternator tends to widen due to the fixed losses associated with the turbine. However the decrease in the efficiency of a constant speed steam turbine at part load is considerably less than with a speed governed turbine. The marine boiler can burn the lowest grades of marine fuel whereas it is only quite recently that small diesel engines for electrical power generation have been developed to burn heavy fuel oil. With the present price of marine diesel oil being approximately 1.6 times that of heavy fuel oil the operating costs are modified accordingly.

There are many medium speed oil engines suitable for marine electrical generation service in the power range 500-2,000 kW. The majority of manufacturers of medium speed oil engine generating sets are situated in Europe and the far east, notably Japan. There are some manufactures in North America which supply to home markets but these engines are only popular with North American owners and do not occupy a large share of the total market for diesel engines. Table 5.2 summarises the principal particulars of available European and North American designed engines.

Engines used for electric power generation are normally four stroke trunk piston engines directly coupled to the alternator and operated at 720, 900 or 1,200 r.p.m., although large machines may be 450 r.p.m. These engines are considerably smaller than the main propulsion engines and have a lower thermal efficiency, partly related to the operating speed. All of the engines can burn diesel oil but only some are capable of burning intermediate and residual fuel oil. Table 5.2 shows the fuel capability of the range of engines presently available.

At the present time the most common fuel consumed by auxiliary engines is distillate oil (MDO), primarily due to the lack of heavy oil burning auxiliary engines. Most vessels which are being built or ordered now are being fitted with heavy oil burning auxiliary engines as more manufacturers are able to offer suitable machines and owners gain operating experience.

Due to the comparatively high cost of fuel, the engine operating costs are the dominant factor in assessment of life cycle cost. The cost of diesel engines suitable for burning heavy fuel is higher than for distillate fuel, principally due to their more sophisticated fuel system, high grade materials and the present speed limitation of 720 r.p.m. Due to the limitation of engine speed brought about by the slower burning heavy fuel the cost of the alternator is also higher, eg., a machine operating at 720 r.p.m. would require a 10 pole alternator for 60 Hz while an 1,800 r.p.m. machine would require a 4 pole alternator. Figure 5.1 shows the cost of alternators versus operating speed.

TABLE 5.1

OPERATIONAL PARAMETERS OF SHIPBOARD ELECTRICAL POWER GENERATION SYSTEMS

	TURBO-ALTERNATOR SET			OIL ENGINE ALTERNATOR SET		
	Steam Ship	Motor Ship		Steam or Motor Ship		
	Main Boiler	Auxiliary Fired Boiler	Waste Heat Boiler			
Fuel	Residual	IF380	-	MDO	IF180	IF380
Lower Calorific Value (LCV) MJ/kg	39.7	40.0	-	41.2	40.2	40.0
Fuel Price $US/Tonne (1)	166	170	-	276	178	170
Efficiency %						
. Primemover (thermal)	22.3	17.2	18.5	37.4	37.4	37.4
. Alternator	94	94	94	94	94	94
. Alternator Set	21.0	16.2	17.4	-	-	-
. Boiler	96	95	97	-	-	-
. System Overall	20.2	15.4	16.9	35.2	35.2	35.2
Equivalent Fuel Rate gms/kW hr (on LCV)	449	584	-	248	254	256
Electrical Energy Cost U.S. cent/kW hr	7.4	9.7	(0)	6.8	4.5	4.4

Note: (1) Averaged Canadian Bunker Rates August 1983

TABLE 5.2

AUXILIARY ENGINE PERFORMANCE DATA

Manufacturer	bmep bar	Speed r.p.m.	Output kW/cyl.	SFC gm/kWhr Guarantee	Fuel Viscosity Limit cSt at 50°C
Allen E1	13.8	720	116	217	420
	13.8	900	144	220	420
MAN E2	14.15	900	81	218	240
	17.9	900	180	208	380
MAK E3	18.0	720	165	209	180
	17.4	900	165	206	180
Sulzer E4	15.84	720	140	211	180
	16.30	900	180	214	180
Werkspoor E5	19.4	900	132	216	200
	17.5	1200	160	216	100
G.M. + E.M.D. NA1	16.2	720	102	239	30
	17.6	900	140	231	30
Fairbanks Morse NA2	-	720	208	219	MDO
	-	900	287	220	MDO
Caterpillar NA3	12.1	1200	52	239	MDO
	10.6	1800	69	248	MDO

Note: Based on fuel with a calorific value of 42700 kJ/kg and ISO standard reference conditions. Add 3% to obtain engine manufacturer's warranty rates.

E - European Design, NA - North American Design

5.2.3 Main Engine Driven Alternators

The advantages of generating electrical power by the main engine are:

a) The main engine has a higher thermal efficiency than the smaller prime movers normally used for electrical power generation.
b) The main engine has the capability to burn heavier and therefore cheaper grades of fuel than the smaller auxiliary engines.
c) It may be possible to eliminate one of the auxiliary alternators.

The electrical power requirement differs widely from ship to ship depending upon operating conditions at sea, in port, during ballasting, cargo operations and maneouvring. Therefore the shaft alternator should be dimensioned to be able to meet the base load electrical requirement when at sea. These alternators need not be able to cope with short term peak loads. In such situations parallel running with separate alternator sets can be effected. For ships requiring extremes of electrical power, for example, self-unloading bulk carriers or ferries, it may be prudent to install a large alternator solely for this purpose.

For constant frequency, the alternator prime mover must run at constant speed. This can be obtained by the use of controllable pitch propellers. However, with fixed pitch propellers, constant speed operation is not possible under all operating conditions. Propulsion plants employing these two types of propellers require different shaft alternator systems. Ships fitted with a CP propeller can operate at constant speed regardless of load and drive a synchronous generator which is able to provide the required "bus bar" frequency and can be coupled directly to the ship's electrical network. For ships fitted with a fixed pitch propeller, the electrical frequency from the synchronous generator must be modified by a frequency converter to provide a constant frequency to the ship's electrical network.

Alternators are constant speed machines designed to supply constant frequency and the tolerance of ship board electrical equipment to variation of frequency is often only ± 2.5 Hz from the design frequency (50 or 60 Hz). Alternators can be coupled directly to an engine at its free end, via gears from the shaft, incorporated directly into the shafting system, via a power take off from the gearbox if fitted, and from the engine camshaft drive system of the Sulzer RTA slow speed engine.

Alternators driven by slow speed marine engines have not been popular for two reasons.

- Slow speed engines are invariably fitted to a fixed pitch propeller and therefore are not regarded as constant speed machines.
- The physical size and therefore cost of a generator fitted to a slow speed engine is large unless gearing is incorporated.

An alternator designed to be directly driven by a slow speed engine at 120 r.p.m. would require 60 poles, and a 90 r.p.m. engine would require 80 poles. These alternators would be excessively large and expensive. Alternators must therefore be driven through gears, and if flexibility of engine operation is to be maintained, a fixed pitch propeller system would require a frequency convertor to allow a constant frequency to be supplied for all modes of operation. Although these systems have occasionally been fitted, they are costly. Recently, Sulzer Brothers have introduced a slow speed engine which has the provision to drive an alternator from a power take off from the camshaft through gears. Although this system permits the alternator to be driven at high speed and thus reduces the number of poles required, the engine must still operate at constant speed if constant supply frequency is to be maintained.

The electrical generation systems which are available and are used at the present time are shown schematically in Figure 5.2 [43, 44]. The most common system fitted to medium speed engines is an alternator driven by a power take off from the reduction gear. This system is also applicable to steam turbine machinery.

- Option 1 requires the alternator to be driven from the front end of the main engine and is applicable to medium and slow speed oil engines.

- Option 2 requires the alternator to be driven from the front end of the main engine through a step up reduction gear to increase the alternator speed and permit a smaller and less expensive alternator to be fitted. This option is suitable for both medium and slow speed oil engines.

- Option 3 requires an alternator to be fitted to the intermediate propeller shafting of a direct drive slow speed oil engine installation.

- Option 4 requires an alternator to be driven by an auxiliary step up gear provided on the slow speed shafting. This option is suitable for all marine systems.

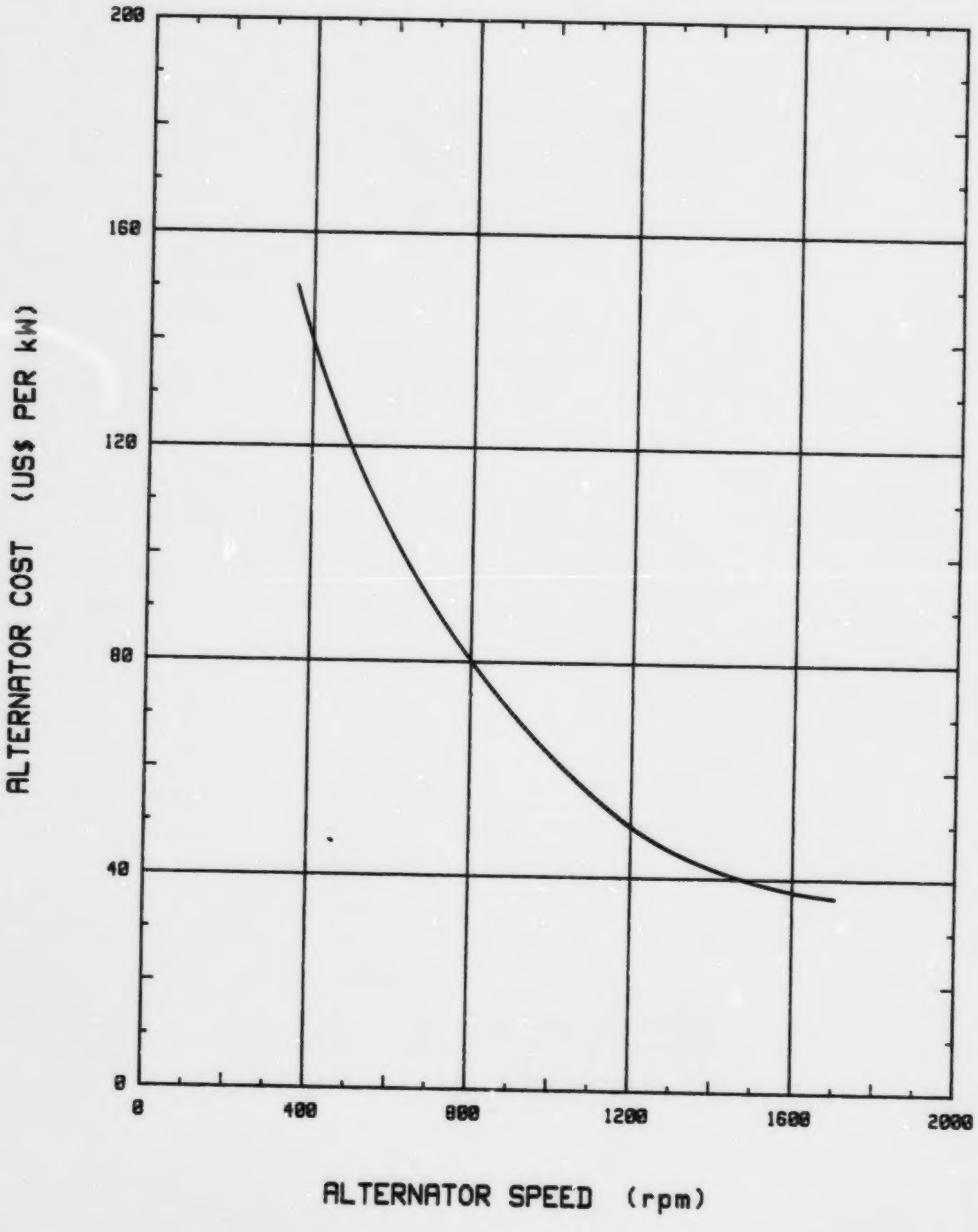
- Option 5 is the most common method in use at present and although adopted mainly for medium speed oil engine installations, is equally applicable for all installations which have a main reducrion gearbox. This system requires a power take off from the gearbox to drive the alternator. The gear ratio of the power take off can be selected to provide any output speed.

The equipment for these five options will depend on whether the propeller is fixed or controllable pitch.

If a "constant frequency" supply is required from an alternator fitted to a variable speed prime mover, a frequency converter or a device to maintain constant speed of the alternator must be fitted. At present, three systems are available:

FIGURE 5.1

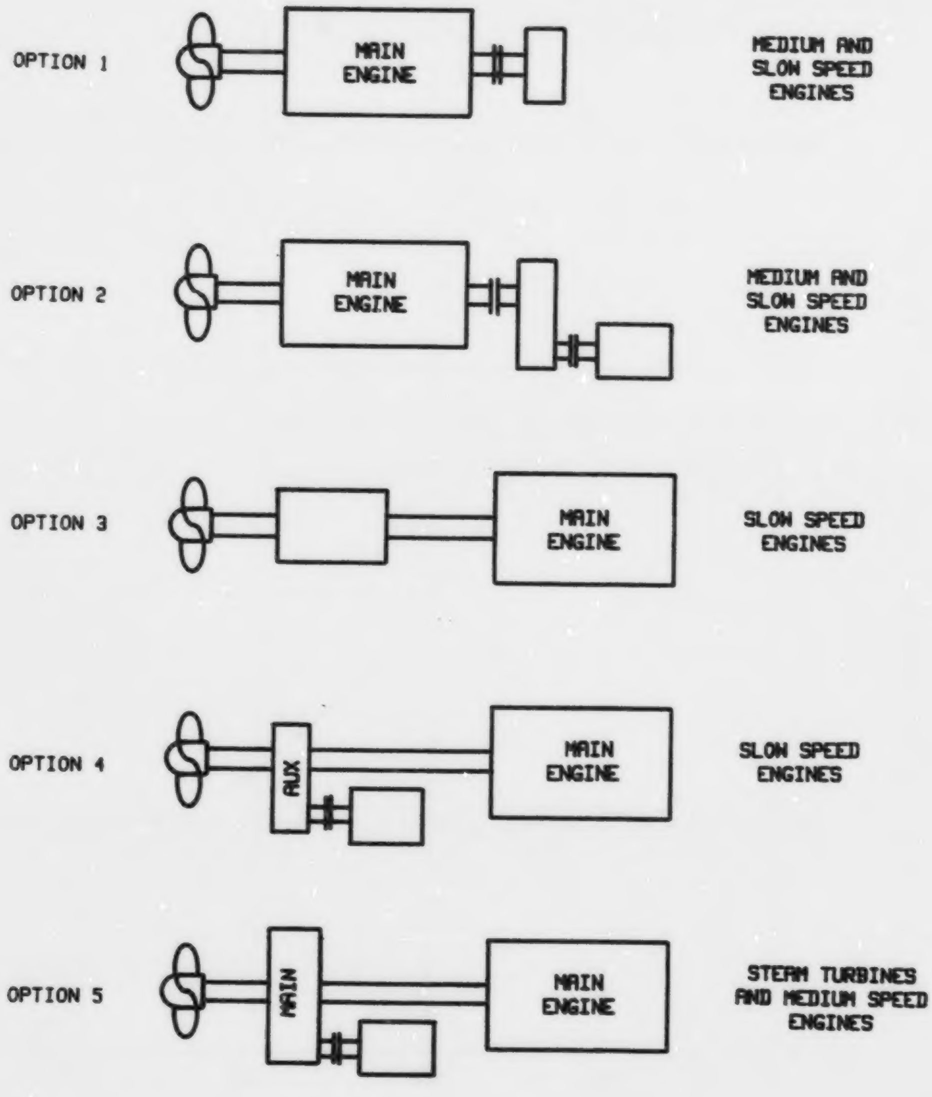
VARIATION OF ALTERNATOR UNIT COST WITH SPEED



* 1983 US Dollars

FIGURE 5.2

MAIN ENGINE POWER TAKE OFF DRIVE SCHEMATICS


OPTION 1
MAIN ENGINE
MEDIUM AND SLOW SPEED ENGINES
OPTION 2
MAIN ENGINE
MEDIUM AND SLOW SPEED ENGINES
OPTION 3
MAIN ENGINE
SLOW SPEED ENGINES
OPTION 4
AUX
MAIN ENGINE
SLOW SPEED ENGINES
OPTION 5
MAIN
MAIN ENGINE
STEAM TURBINES AND MEDIUM SPEED ENGINES

[43, 48]

- static frequency conversion
- rotary frequency conversion
- constant speed hydraulic gear

A static frequency converter is relatively new and has been made posssible by developments in the field of semi-conductors. The shaft alternator produces a three phase alternating current of varying frequency depending on the speed of the main engine. The alternating current is rectified and inverted by a thyristor control, set to "fire" at the required frequency of the output. As the frequency converter has a DC intermediate link, no reactive effect can be supplied to the electric mains. To remedy this a synchronous condenser is incorporated into the electrical system for power factor correction. The sychronous condenser may be a separate electrical machine or the alternator part of a diesel generator set already fitted. Figure 5.3 shows a schematic arrangement of this system [45].

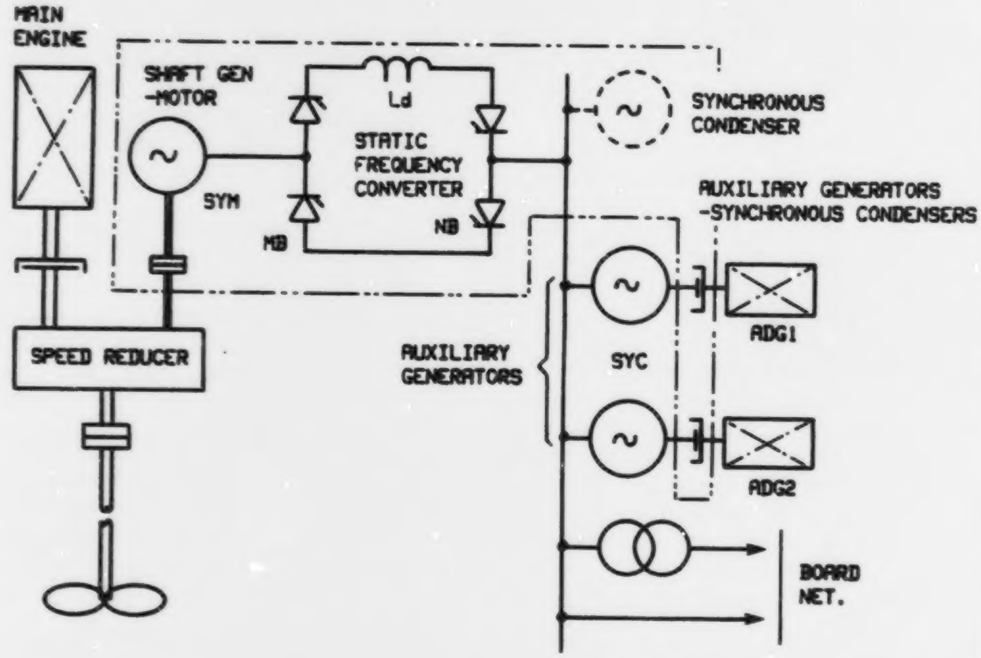
In the rotary converter system the variable frequency supply from the main engine driven alternator is rectified and supplied to a constant speed DC motor which drives a three phase AC alternator. The AC output is of constant frequency. An alternative arrangement uses a DC generator driven by the main engine which directly connects electrically with a DC motor. The DC motor again drives an AC alternator at constant speed. With either of these systems it is possible to use the alternator of an auxiliary set and thus save the cost of one electrical machine. Figure 5.4 shows two schematic diagrams of these rotary converter systems [43, 46].

The third system for providing constant frequency from the main engine driven alternator systems is through hydraulic-gearing with the so called constant speed generator drive (CSGD). This system provides a constant output speed from a variable input speed. Figure 5.5 shows a schematic arrangement of this system. The constant speed AC generator is connected to one of three elements of an epicyclic gear system. The main engine drive is connected to one of the remaining two elements and the third element is regulated to give the required constant speed of the generator. Speed control of the third element is achieved using two hydraulic units, one coupled to the main engine drive and the other to the third element of the epicyclic gear. Variation in main engine speed alters the outputs from the variable and fixed displacement hydraulic units which in turn alters the relative rotational speeds within the gear to produce a constant speed to the shaft generator [47].

The limitations of shaft alternator systems must be viewed in relation to the power profile of the vessel. Normally shaft alternators are fitted on ocean going vessels trading on extended sea routes and can use the alternator for the total sea load without supplementary power from auxiliary generating sets. When it is required to operate the shaft alternator system in parallel with auxiliary systems this can be achieved without difficulty where frequency convertor units are fitted to the shaft generator. However, when frequency convertors are not used, then the characteristics of the speed control of the main engine and auxiliary system must be matched. Normally the governor system for the auxiliary engine is provided with a speed droop characteristic whereas the speed control for the main engine is quite different. The output load of the shaft alternator constitutes a

FIGURE 5.3

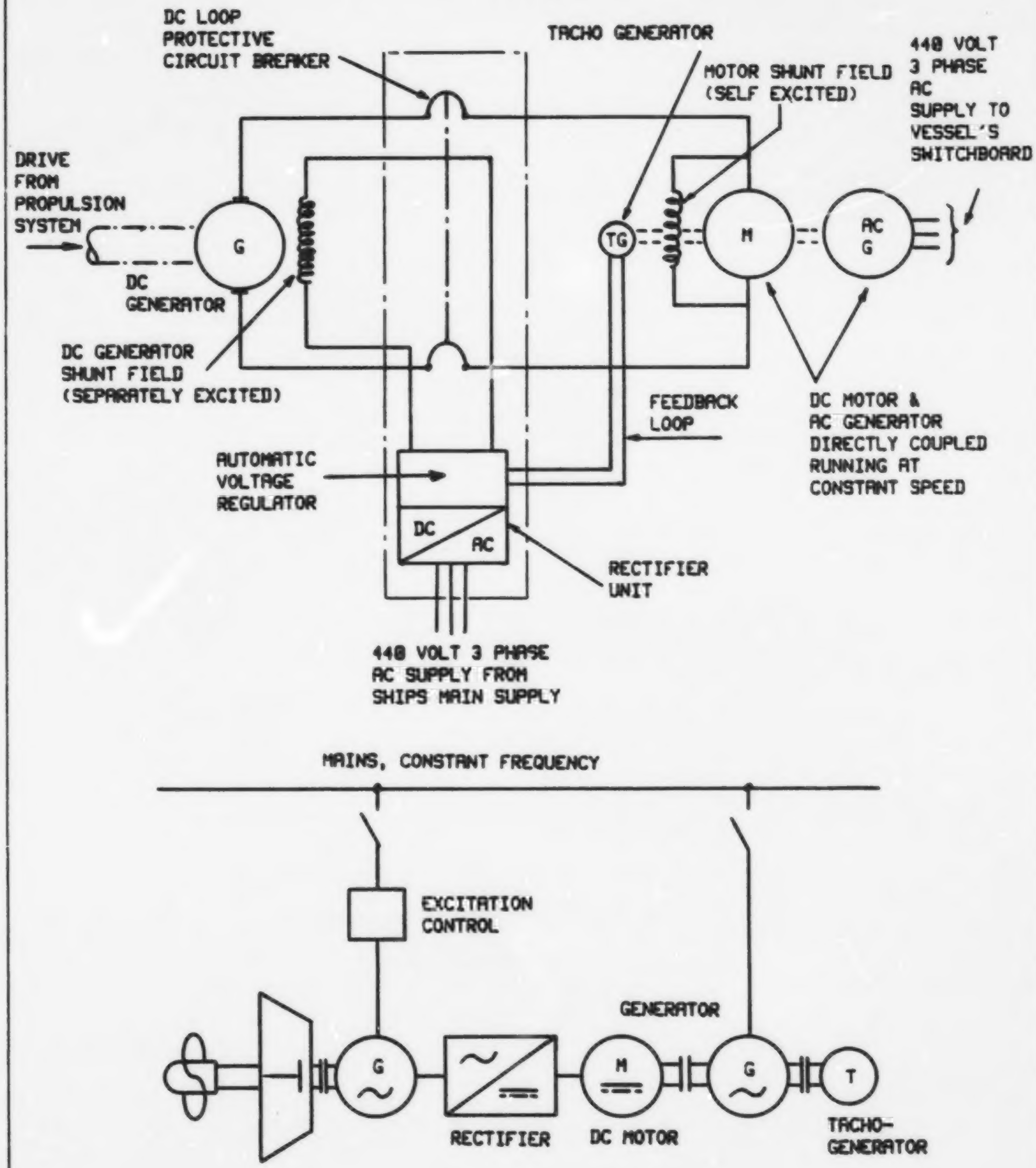
SCHEMATIC DIAGRAM OF MAIN ENGINE DRIVEN ALTERNATOR WITH STATIC FREQUENCY CONVERTER


MAIN ENGINE
SHAFT GEN -MOTOR
Ld
STATIC FREQUENCY CONVERTER
SYM
MB
NB
SYNCHRONOUS CONDENSER
AUXILIARY GENERATORS -SYNCHRONOUS CONDENSERS
SPEED REDUCER
AUXILIARY GENERATORS
SYC
ADG1
ADG2
BOARD NET.

[45]

FIGURE 5.4

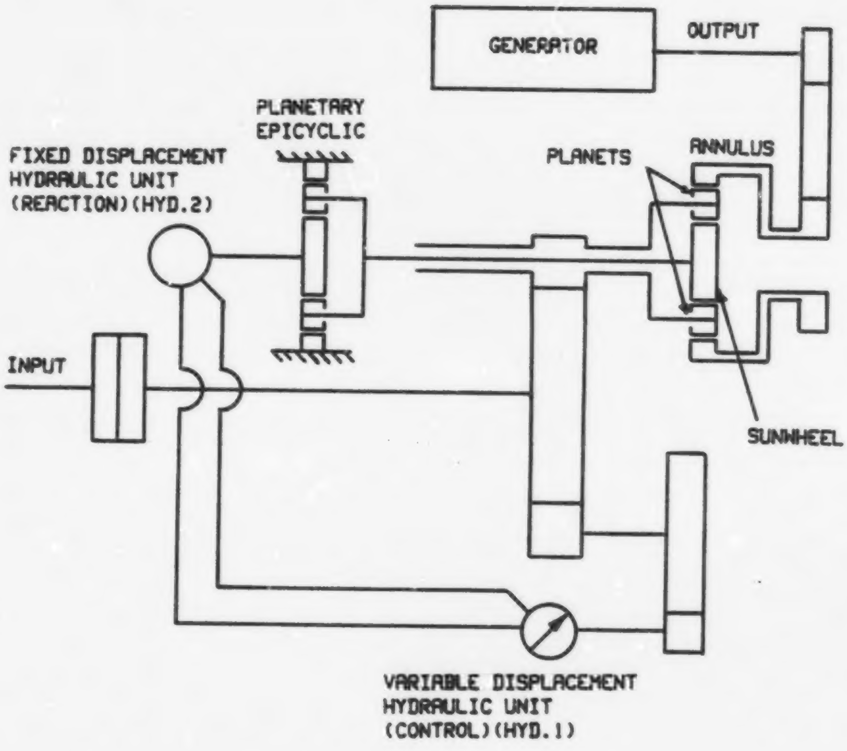
SCHEMATIC DIAGRAM OF MAIN ENGINE DRIVEN ALTERNATOR WITH ROTARY CONVERTER


DC LOOP PROTECTIVE CIRCUIT BREAKER
TACHO GENERATOR
440 VOLT 3 PHASE AC SUPPLY TO VESSEL'S SWITCHBOARD
MOTOR SHUNT FIELD (SELF EXCITED)
DRIVE FROM PROPULSION SYSTEM
G
DC GENERATOR
TG
M
AC G
DC GENERATOR SHUNT FIELD (SEPARATELY EXCITED)
FEEDBACK LOOP
DC MOTOR & AC GENERATOR DIRECTLY COUPLED RUNNING AT CONSTANT SPEED
AUTOMATIC VOLTAGE REGULATOR
DC
AC
RECTIFIER UNIT
440 VOLT 3 PHASE AC SUPPLY FROM SHIPS MAIN SUPPLY
MAINS, CONSTANT FREQUENCY
EXCITATION CONTROL
GENERATOR
G
M
G
T
RECTIFIER
DC MOTOR
TACHO-GENERATOR

[43, 46]

FIGURE 5.5

SCHEMATIC DIAGRAM OF MAIN ENGINE DRIVEN ALTERNATOR WITH CONSTANT SPEED GENERATOR DRIVE


GENERATOR
OUTPUT
PLANETARY EPICYCLIC
FIXED DISPLACEMENT HYDRAULIC UNIT (REACTION)(HYD.2)
PLANETS
ANNULUS
INPUT
SUNWHEEL
VARIABLE DISPLACEMENT HYDRAULIC UNIT (CONTROL)(HYD.1)

[47]

relatively small part of the main engine load and consequently the maximum torque of the alternator does not influence the main engine to maintain the electrically synchronous position. Stable parallel operation of shaft driven alternators is therefore not normally possible unless the governor and the load response characteristics of the systems are compatible.

5.2.4 Capital and Energy Costs

To evaluate shaft alternator systems the capital cost should be compared with the more conventional systems. Figures 5.6, 5.7 and 5.8 give estimated cost of alternators, step up gearboxes and frequency conversion systems. Figure 5.9 shows the capital cost of auxiliary engine/alternators sets which may be used for comparison. Figures 5.10, 5.11, 5.12 and 5.13, based on Figures 5.6 to 5.8 and data obtained from producers and references [43, 47, 48], show the combined equipment costs for the options shown in Figure 5.2. For each option the difference between fixed pitch and CP propeller installations has been shown and it has been assumed that no frequency conversion equipment is provided for the CP propeller system.

From these figures it may be seen that the installations with fixed pitch propellers are not competitive with the auxiliary oil engine alternator sets on capital cost. This information is vital in assessing the likelihood of adopting such systems.

With regard to the efficiency and hence the energy consumption of the systems Table 5.3 shows the combined system efficiency made up from the component efficiencies of the different systems. It has been assumed that the gear/step up gear efficiencies are 97.8% and 98%. Alternator efficiencies vary between 88% and 94% depending on the rotational speed. Static frequency coversion systems are 94.5% efficient, rotary frequency conversion systems are 86% efficient anad the constant speed gear hydraulic drive system varies between 93.5% and 98% depending on the speed variation of the main engine. These efficiencies have been taken from current literature [43, 47] and from manaufacturers. Depending on the system used, the electrical generation efficiency varies between 79% and 92%. If main engine efficiency is also considered the efficiency varies between 23% and 42%. As would be expected the highest efficiencies are achieved with CP propellers where frequency conversion systems are unnecessary. Also included in this table is the cost of producing electrical energy based upon equivalent fuel rates from the main engine, the different types of fuel as shown and the respective fuel prices.

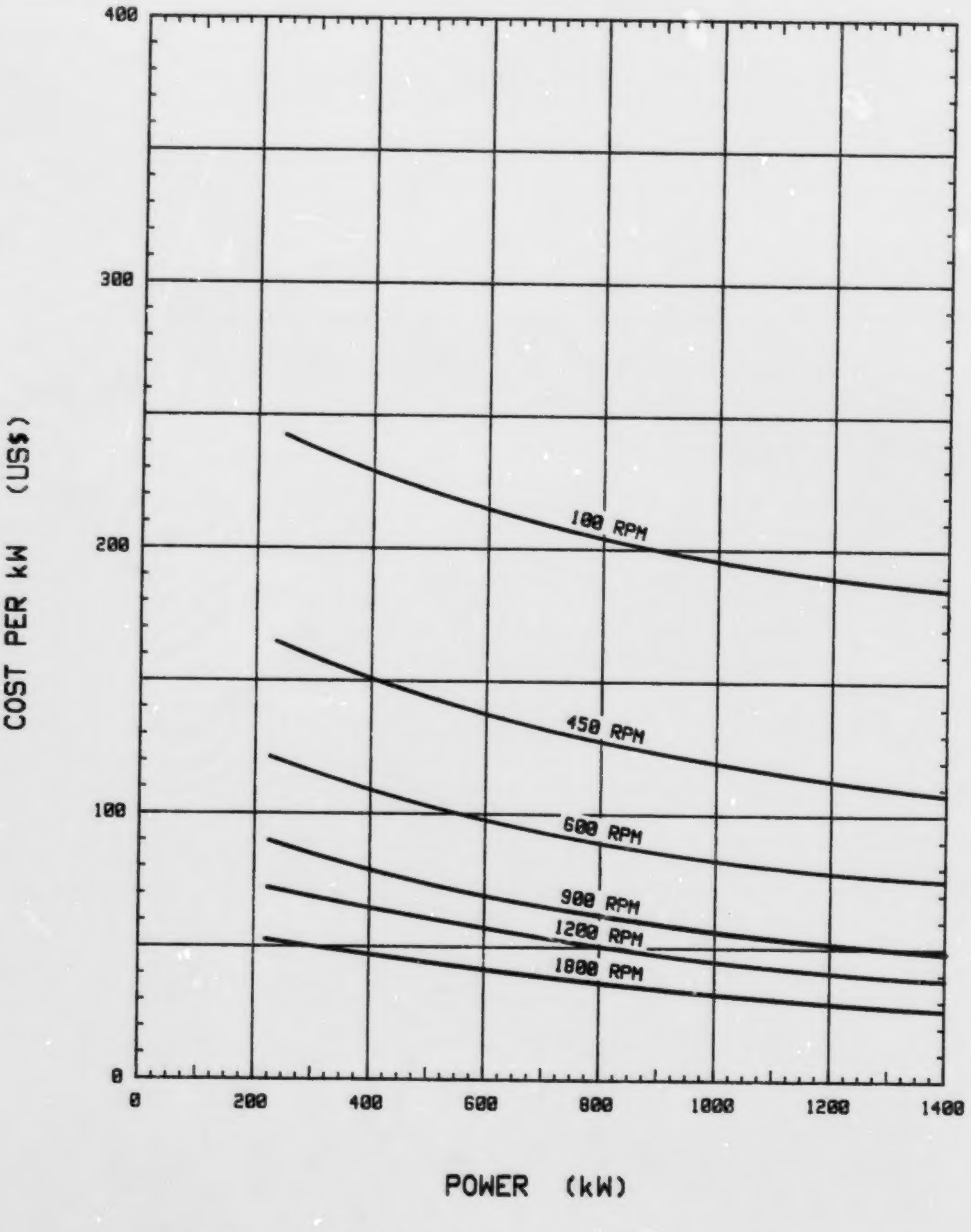
These energy costs must be translated into annual operating costs, making them highly dependent on the operating profile of the vessel.

5.3 Heating Systems

The heating demand on board ship, is to preheat heavy or residual fuel bunkers, to provide accommodation and space heating, galley and hot water service and on some vessels for heating the cargoes. Heat is distributed from its source electrically, by steam or by the circulation of hot water or thermal fluids.

FIGURE 5.6

CAPITAL COST OF ALTERNATOR VERSUS RPM



* Multiply cost by 1.5 to obtain value for variable speed alternator with full load at part load speed

FIGURE 5.7

CAPITAL COST OF GEAR BOX

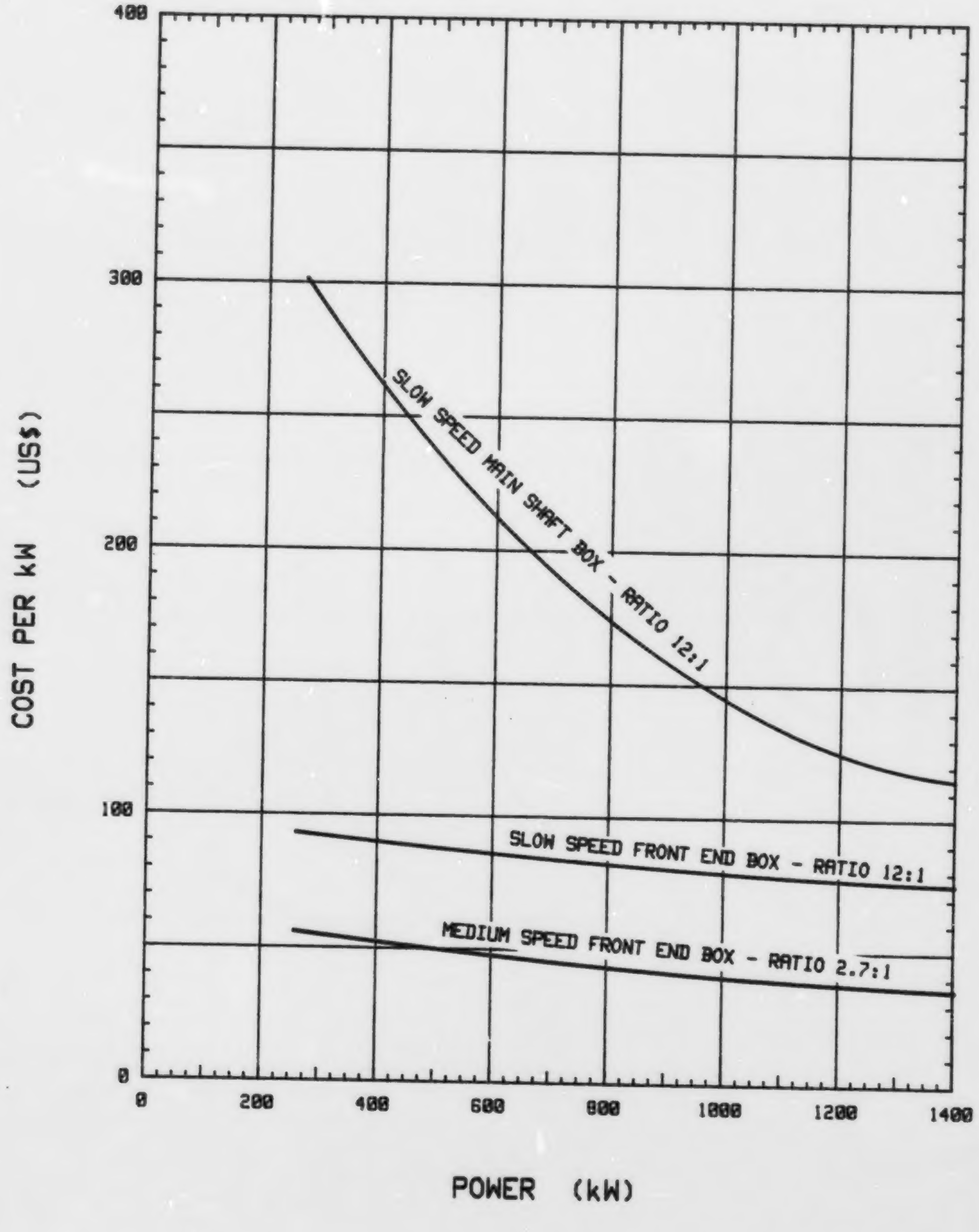


FIGURE 5.8

CAPITAL COST OF FREQUENCY CONVERTER

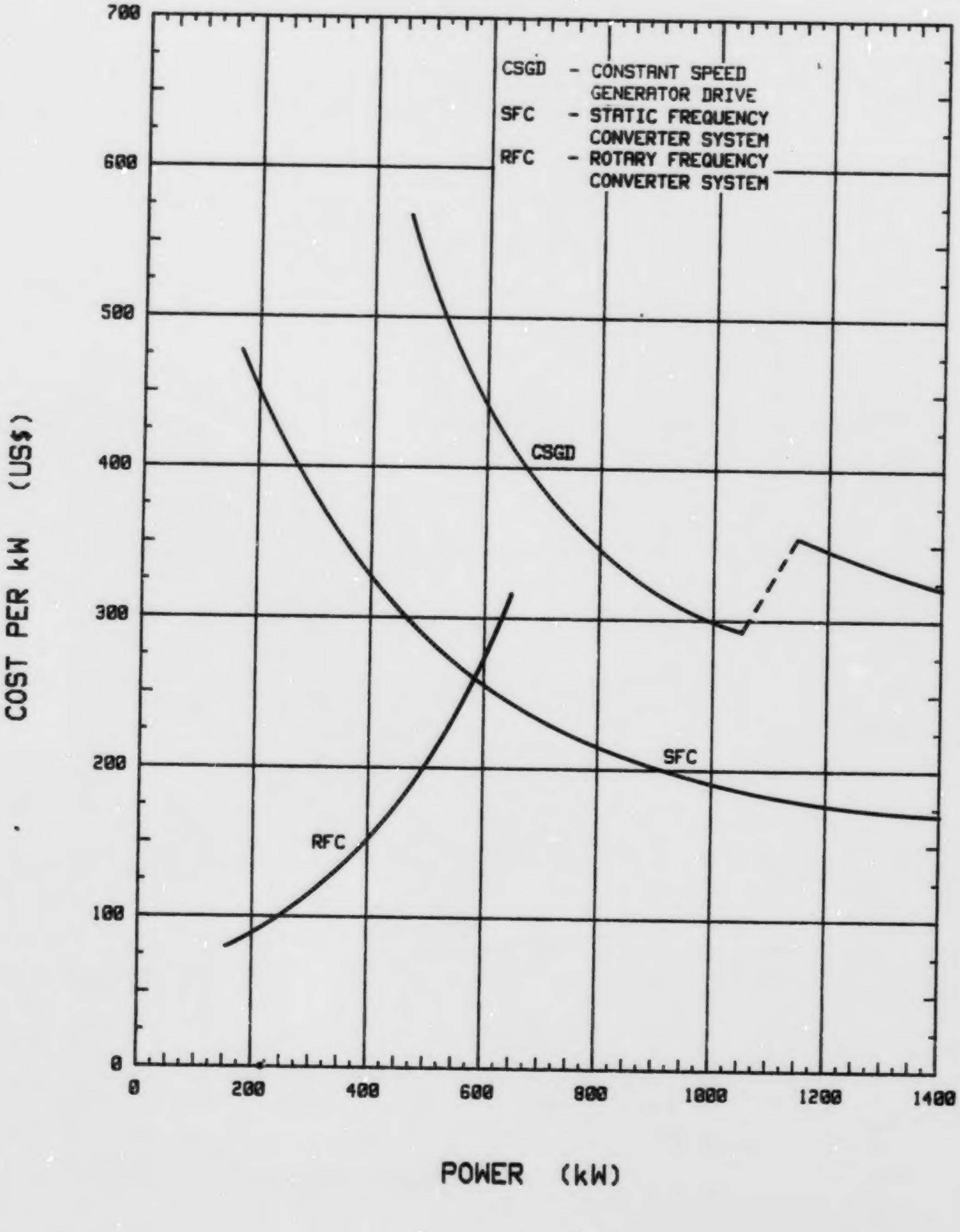


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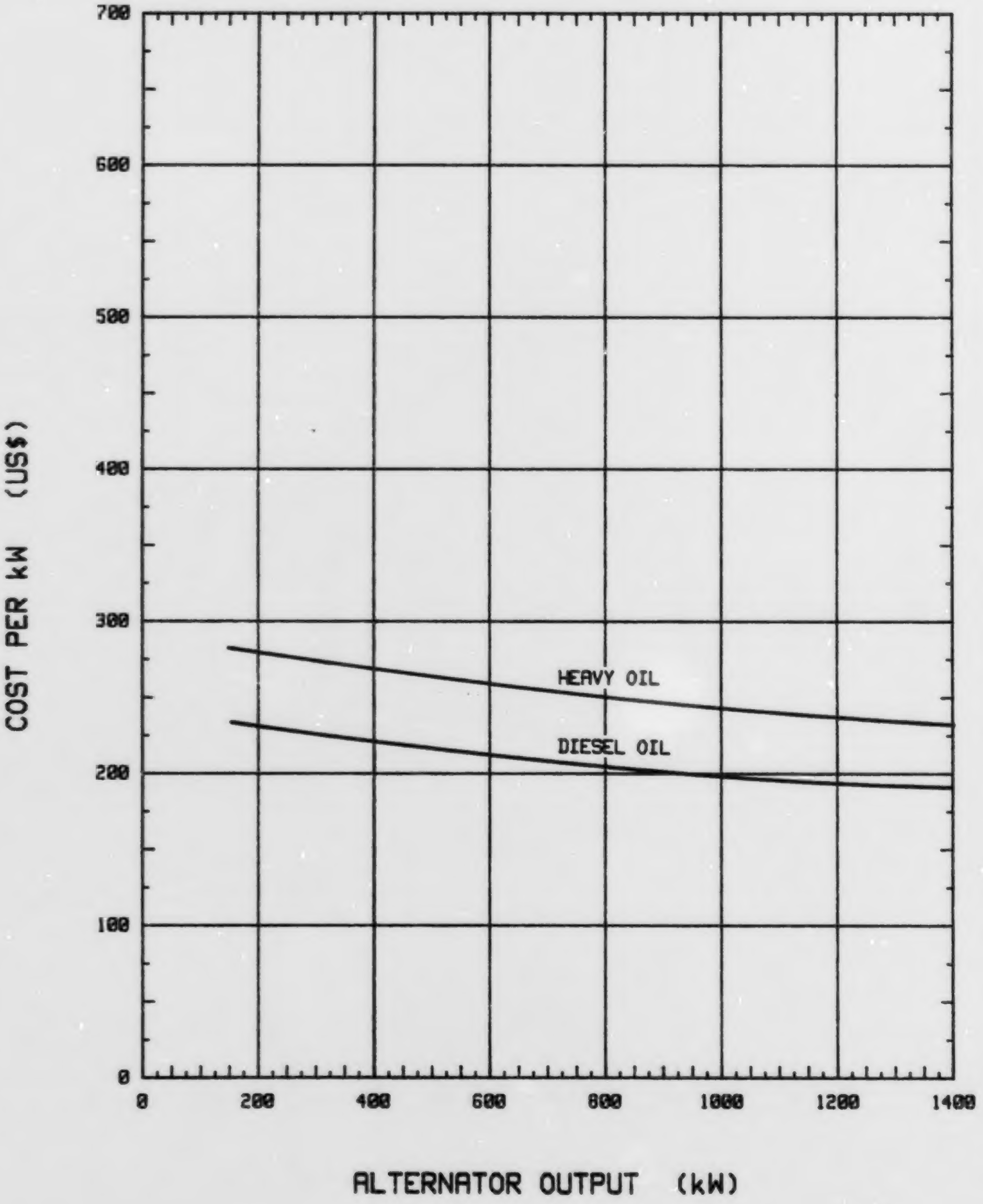
FIGURE 5.9

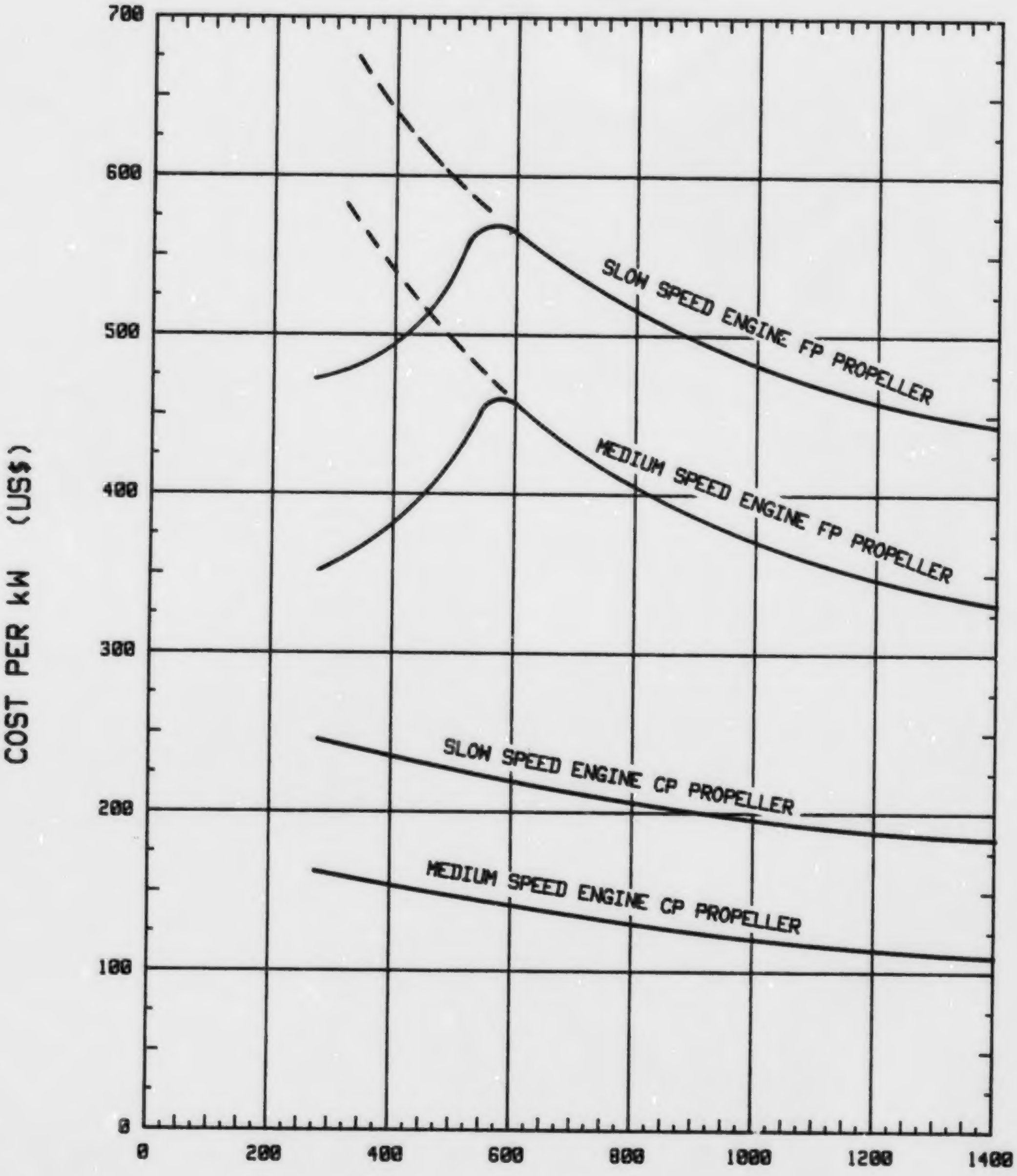
CAPITAL COST OF ALTERNATOR SYSTEM



* Auxiliary oil engine/alterator set

FIGURE 5.10

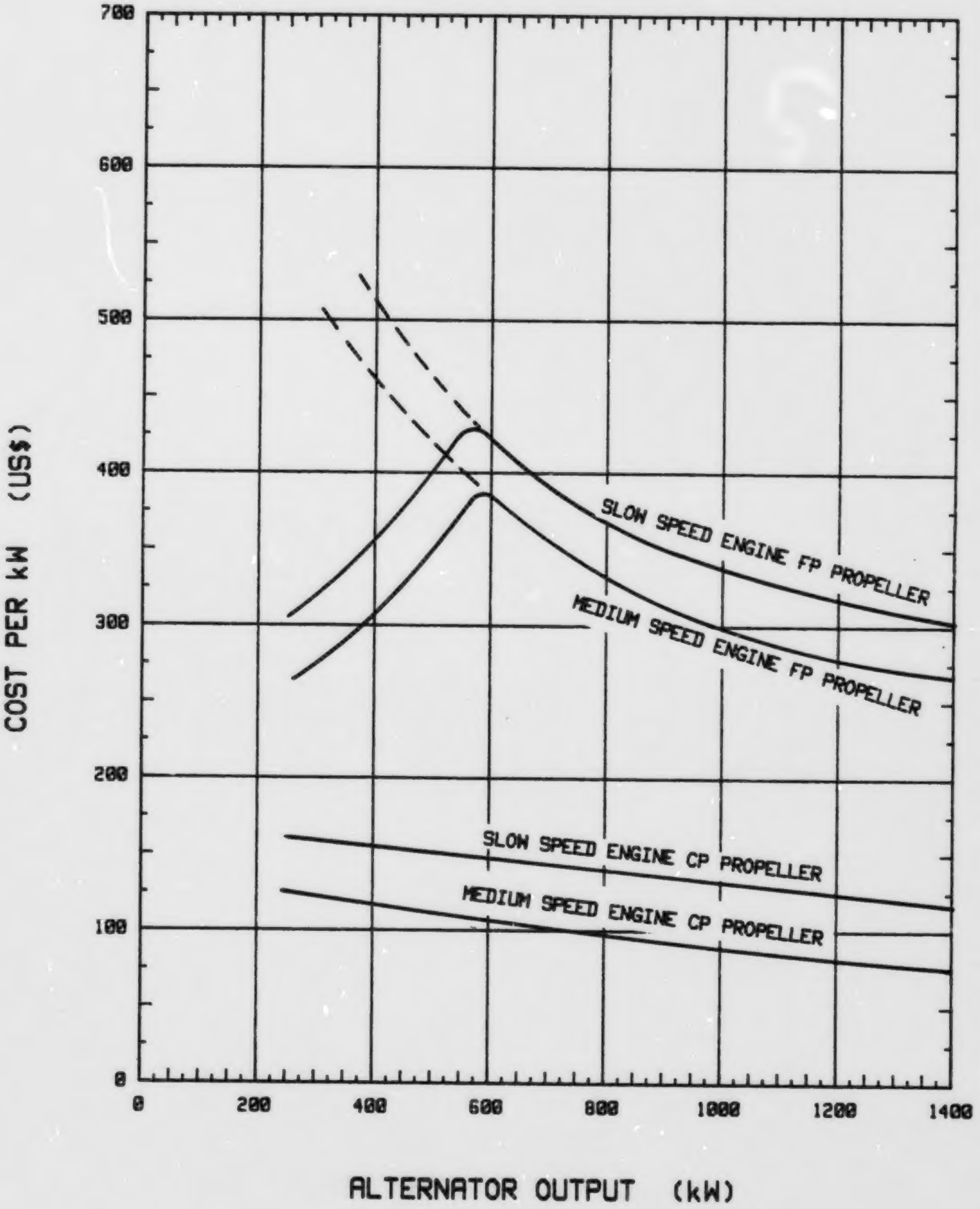
CAPITAL COST OF ALTERNATOR SYSTEM
OPTION ONE



* Main engine front end driven alternator, direct drive

FIGURE 5.11

CAPITAL COST OF ALTERNATOR SYSTEM
OPTION TWO



* Main engine front end driven alternator with step-up to 1200 RPM

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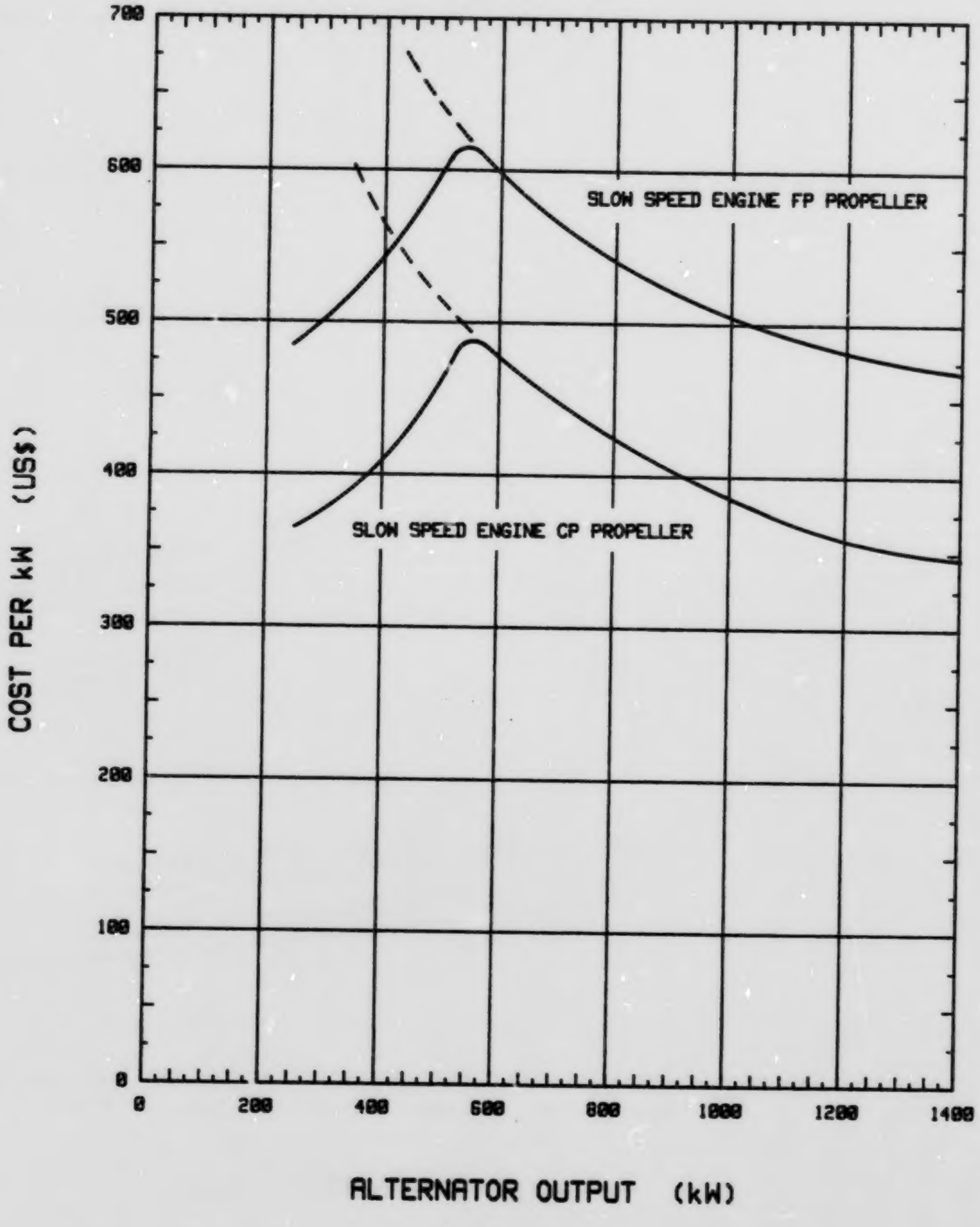
FIGURE 5.12

CAPITAL COST OF ALTERNATOR SYSTEM
OPTION THREE



* Main engine alternator on slow speed shaft line

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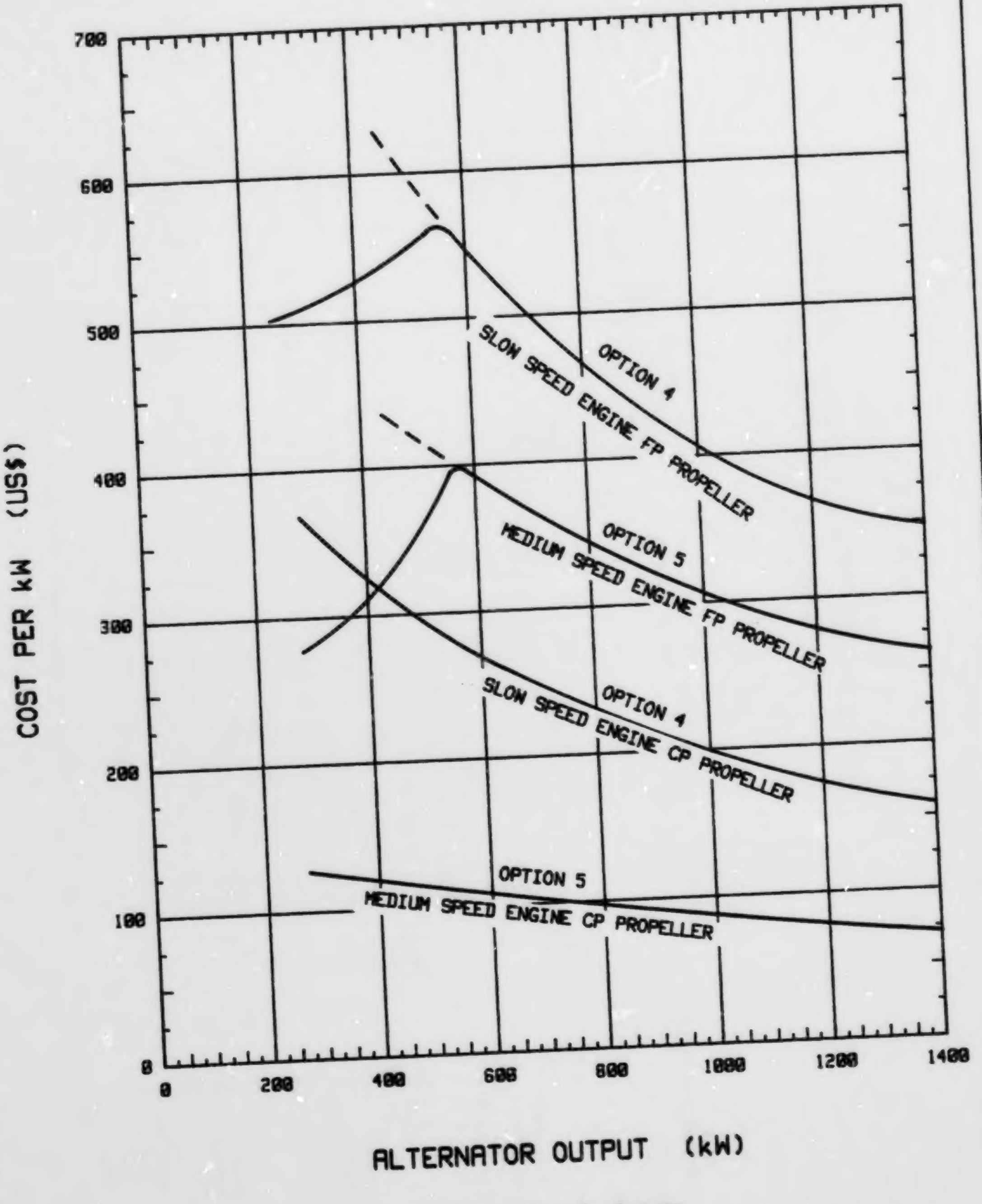
FIGURE 5.13

CAPITAL COST OF ALTERNATOR SYSTEM
OPTIONS FOUR & FIVE



* Main engine alternator driven through step-up gear on main shafting at 1200 RPM

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TABLE 5.3

EFFICIENCIES OF MAIN ENGINE DRIVEN ALTERNATOR SYSTEMS (%)

MAIN ENGINE	PROPELLER	DRIVE		CONVERTER			ALTERNATOR	SYSTEM		FUEL LCV MJ/Kg AND PRICE $U.S./ton	EQUIV. FUEL RATE ON LCV gms/kW/hr	ENERGY COST U.S.c/kW/hr
		DIRECT	GEAR	SFC	RFC	CSGD		WITHOUT MAIN ENGINE	WITH MAIN ENGINE			
	FP	99.7	-	94.5	-	-	88	82.9	40.0		225	3.83
	FP	99.7	-	-	86	-	88	75.5	35.5		254	4.32
	FP	99.7	-	-	-	93.5-98	88	82.0-86.0	38.5-40.4		234-233	3.98-3.79
Slow Speed	FP	-	98.0	94.5	-	-	94	87.1	40.9	40.0/	220	3.74
Diesel	FP	-	98.0	-	86	-	94	79.2	37.2	170	242	4.11
(Long	FP	-	98.0	-	-	93.5-98	94	86.1-90.2	40.5-42.4	(IF380)	222-212	3.77
Stroke) 47	CP	99.7	-	-	-	-	88	87.7	41.2		218	3.60
	CP	-	98.0	-	-	-	94	92.1	43.3		208	3.54
	FP	99.5	-	94.5	-	-	92	86.5	37.2		242	4.60
	FP	99.5	-	-	86	-	92	78.7	33.9		265	5.04
	FP	99.5	-	-	-	93.5-98	92	85.6-89.7	36.8-38.5		245-234	4.65-4.45
Medium	FP	-	97.8	94.5	-	-	94	86.9	37.4	40.0	241	4.58
Speed	FP	-	97.8	-	86	-	94	79.1	34.0	190	265	5.04
Diesel 43	FP	-	97.8	-	-	93.5-98	94	85.9-90.1	36.9-38.7	(IF180)	244-233	4.64-4.43
	CP	-	-	-	-	-	92	92.0	39.6		227	4.31
	CP	-	97.8	-	-	-	94	91.9	39.5		228	4.33
	FP	-	97.8	94.5	-	-	94	86.9	25.2		360	5.98
Steam	FP	-	97.8	-	86	-	94	79.1	22.9	39.7/	396	6.57
Turbine	FP	-	97.8	-	-	93.5-98	94	86.0	24.9	166	364	6.04
29	CP	-	97.8	-	-	-	94	91.9	26.7	Residual	340	5.64
- Comparison Auxiliary Engine/Alternator Set												
37.4	-	-	-	-	-	-	94	35.2		MDO/275	248	6.82
37.4	-	-	-	-	-	-	94	35.2		IF180	254	4.82
37.4	-	-	-	-	-	-	94	35.2		IF380	256	4.35

Table 5.4 shows the onboard heat users and the systems which can be used to distribute the required heat and methods which may be used to control the heat loss. Each of the heat user systems is categorised by the maximum temperature required. The required temperature will have a significant influence on the type of heat distribution medium that may be employed. In this regard high grade heat refers to temperatures above 100°C, the boiling point of water. The table indicates which of the heat distribution systems are most suitable for each system. Each system is designated for minimum energy usage in each of the demand areas and is compared with present practice.

As previously discussed there is a trend towards increasing the viscosity of residual fuel oils as more of the lightest fractions are extracted. In order for the fuel oil to be pumpable, and to permit the extraneous heavy material and water to settle, the bunker storage spaces and settling tanks must be heated.

Heating of heavy fuel bunkers can represent a substantial proportion of the auxiliary heat load of the vessel. In the case of steam turbine powered vessels the auxiliary heat load is invariably provided by the boilers or indirectly from steam bled from the turbines. Whatever method is used there is a cost incurred as fuel must be consumed to provide the required heat.

The use of positive displacement pumps offer a substantial advantage for bunker transfer duties as these pumps are better suited to pumping high viscosity fluids than centrifugal pumps and thus the storage temperatures in the bunker tanks may be reduced. A reduction of the bunker heating load will substantially reduce the heating costs. In addition, the heating medium pipework should be carefully installed and insulated to prevent loss of heat.

Auxiliary heat supplied from a boiler system using either steam, water or a thermal fluid for the distribution medium is more efficent, with respect to fuel conservation, than the same quantity of heat generated electrically. The capital cost of a smaller electric heating installation is less than for a boiler, but for any system other than the smallest, it is more economical overall to supply the auxiliary heat load from an exhaust or diesel fired boiler system rather than by electric heating.

If the vessel's heating requirements are met by burning marine diesel oil (MDO) then the energy costs increase in proportion to the higher price of MDO, i.e., by a factor of 1.62. If electrical energy is generated by the main propulsion engine, to take advantage of its higher thermal efficiency the equivalent electrical heat cost would reduce to approximately 3.8 U.S. cents per kW for residual fuel at $170 U.S. per tonne.

The cost of heat supplied from a boiler may be regarded as applicable to a steam system, a hot water system or a thermal fluid system, although the operating cost of a steam system may be marginally higher due to less than total condensation of the steam within heating coils of the system. The figures show the boiler plant to be the most economic and energy efficient system but the less than total acceptance of this system by the industry is probably influenced by the increased capital cost, maintenance and complexity compared with heating by electricity.

The use of thermal fluid systems using mineral oils, synthetic hyrdocarbons or diphenyl/diphenyl oxide is interesting since nearly all onboard heating requirements can be met by this system. However, the energy saving benefits from its use, if any, are marginal compared with hot water or steam systems. The higher capital and system operating costs, and the risk of fire from leaks during the heat absorption process in either a direct fired or an exhaust gas boiler appear to contribute to its poor acceptance by shipowners in all but very few specialist applications.

As fuel qualities worsen there will be an increase in energy requirements for fuel oil heating both with respect to direct heat to reduce the viscosity to enable the fuel to be pumped and to raise the temperature sufficently to allow the bunkers to "settle". In addition the fuel must be preheated before the engine to allow atomisation by the fuel injection in the engine cylinder. Hence fuel oil heating requirements will become a larger percentage of the total energy requirements of ships of the future.

5.4 Waste Heat Recovery

Although the thermal efficiency of marine steam turbine installations rarely exceeds 30%, i.e., 70% of all heat is rejected to the sea water or the atmosphere, there is little scope for recovery of this heat. Boiler exhaust gas temperatures at outlet from the boiler are purposely designed to be a minimum for high boiler efficiency. Similarly the turbines are designed to extract the maximum heat from the steam before it is exhausted to the condenser. Although the heat content of the exhausted steam is high, the temperature is low and therefore difficult and expensive to recover. Similarly the heat which is rejected in the boiler exhaust gases is in total large but what is economically recoverable is determined by the dew point of the gases and is already recovered by the air heater.

It is therefore concluded that waste heat recovery from turbine/boiler systems is not practical and the remainder of this section will deal with oil engine installations.

5.4.1 Heat Sources

Oil engines have a thermal efficiency between 40-50% and reject 50-60% of the heat energy supplied in the fuel. This heat is discarded in the exhaust gases and from the engine cylinders, pistons, charge air and lubrication oil to the sea through the cooling system. Figure 5.14 shows the typical energy dissipations for slow speed two stroke and medium speed four stroke engines. The figure is a typical Sankey diagram based on data from producers' catalogues.

The grade of heat dissipated in the various systems fall into two separate categories. Exhaust gas heat is at a relatively high temperature, typically 270-300°C in the slow speed long stroke engines and between 340-370°C for the medium speed engines. The second category, typified by the cooling water system, is of relatively low grade heat with temperatures in the region 70-90°C. However, a few engine manufacturers are now proposing that jacket water temperatures can be increased to 120°C by the use of pressurized systems which will improve the thermal efficiency of the engine system and permit easiser heat recovery from the system.

TABLE 5.4

HEAT DEMAND ON BOARD SHIP

DEMAND	MAX. TEMP OF HEATED MEDIUM (^0C)	GRADE OF HEATING MEDIUM	HEATING MEDIUM OPTIONS STEAM (S) H	 ELECTRIC (E) H	 HOT WATER (HW) L	 THERMAL FLUIDS (TF) H/L	MIN. ENERGY SOURCE	NORMAL PRACTICE	POTENTIAL DESIGN AREAS OF REDUCING ENERGY CONSUMPTION
Engineroom Settling and Service Fuel Tanks	52	H/L	X		X	X	HW	S	Insulation of tanks, efficient steam traps, temp. control, heating coil design
Fuel Oil Purifier Heaters	70-98	H	X	X		X	S	S or E	Temperature control
Fuel Oil Line Heaters	146	H	X	X		X	S	S or E	Temperature control
Lub Oil Tanks	52	H/L	X		X	X	HW	S	Insulation of tanks, temperature control
Lub Oil Purifier Heaters	93	H	X	X		X	S	S or E	Temperature control
Main Engine Water Heater	80	H/L	X		X	X	S	S	Engine insulation
Distiller	100	H/L	X		X	X	HW	HW or S	Performance control, adoption of reverse osmosis units
Sea Chest Cleaning/De-icing	45	H	X		X		HW	S	-
Domestic Water	82	H	X	X	X	X	HW	S or E	Temperature control, domestic water valve system design
Outside Engineroom Air Conditioning Plant	50	H/L	X	X	X	X	HW	S/E/HW	Accommodation layout, insulation & thermostatic control at 18°C
Oil Fuel Bunker Tanks	40	H/L	X		X	X	HW	S	Optimisation of tank numbers, and position in ship. Heating coil design
Galley	-	H		X			E	E	Galley design/ergonomics
Ship Spaces	15	H/L	X	X	X	X	HW	E or S	Space insulation

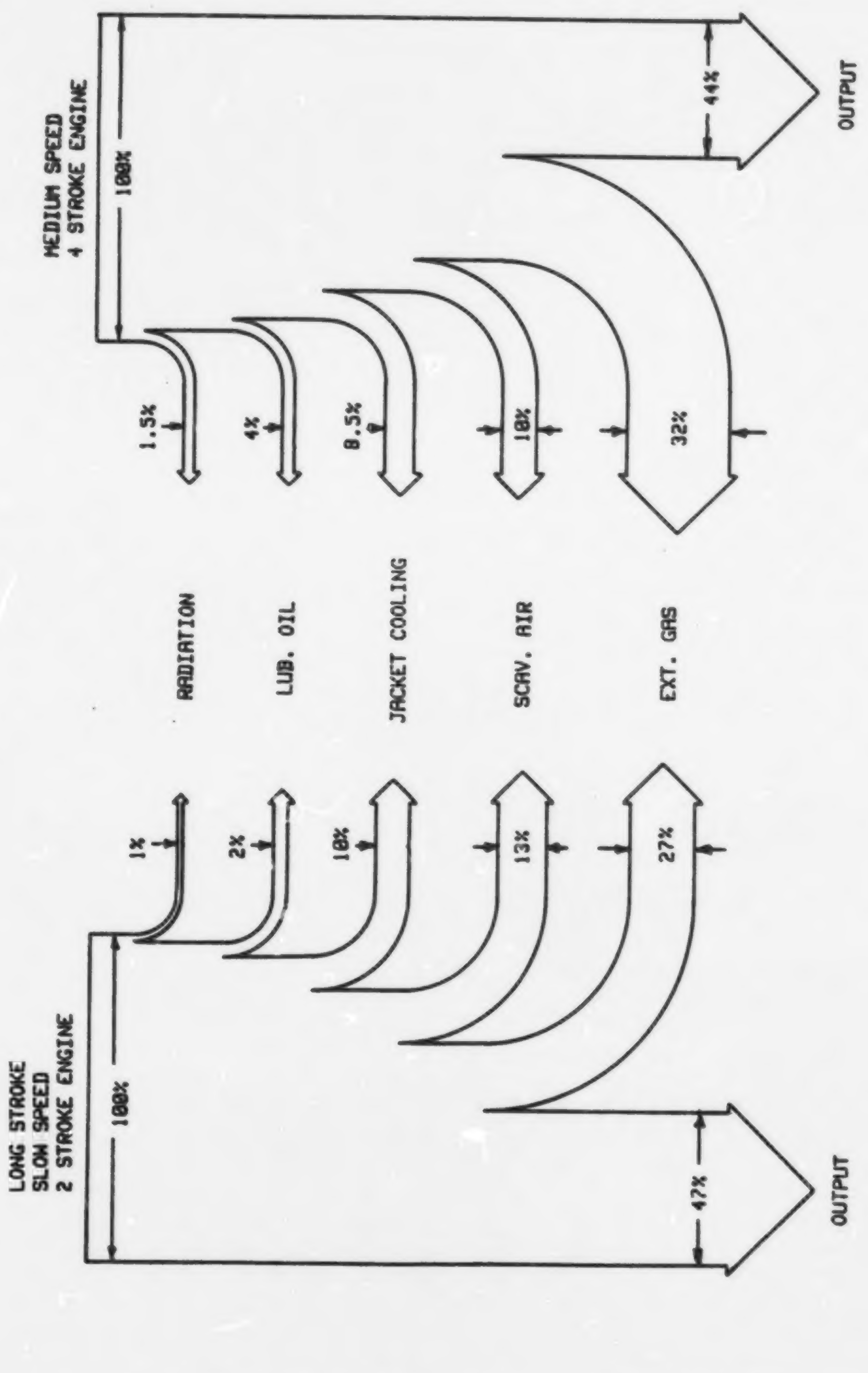
N.B. Minimum energy source assumed that steam or hot water is produced by engine waste heat recovery systems.

H - High Grade Heat ≥ 100°C

L - Low Grade Heat < 100°C

FIGURE 5.14

TYPICAL ENERGY DISSIPATIONS OF DIESEL ENGINES


MEDIUM SPEED
4 STROKE ENGINE
100%
44%
OUTPUT
1.5%
4%
8.5%
10%
32%
RADIATION
LUB. OIL
JACKET COOLING
SCAV. AIR
EXT. GAS
1%
2%
10%
13%
27%
LONG STROKE
SLOW SPEED
2 STROKE ENGINE
100%
47%
OUTPUT

5.4.2 Exhaust Gas

The recovery of the high grade heat in the exhaust of main propulsion diesel engines by the use of composite boilers has been common practice for many years. The following text describes:

- exhaust gas waste heat recovery systems including; the medium and the equipment
- estimates of the recoverable heat, and
- the potential for generation of electric power

a) Recovery Systems

The condition and quantity of the steam at the boiler stop valve is governed by the temperature and mass flow of exhaust gases from the engine. In general the mass flow of exhaust gases from medium speed four stroke engines are slightly higher than for slow speed two stroke engines. The variation in specific exhaust gas mass flow is dependent upon the engine vintage rather than speed or engine type, i.e., 2 stroke or 4 stroke. As engine efficiency has increased so the quantity of exhaust gases has decreased. In the early 1960's engines produced about 10 kg/kW hr of exhaust gases. From published data, an average figure of 8 kg/hr of exhaust gas per engine kilowatt load at MCR conditions have been derived and is used for evaluation.

Engines do not operate continuously at their maximum continuous rating and therefore the part load characteristics are of significance. Figure 5.15 shows the variation of specific exhaust gas quantities with engine load for an intake air temperature of 27°C. The part load flow factor shown in the figure can be applied to the 8 kg/kW hr for operating conditions away from MCR. The data for this figure has been taken from manufacturers' published literature [50, 52].

The data quoted by engine manufacturers for engine performance is normally based upon standard conditions (International Standards Organization Standard 3046/1), air temperature 27°C, cooling water before charge air cooler 27°C, air pressure 1 bar. The continuous service rating (CSR) is based upon 85% MCR. Marine propulsion engines are also given a rating named maximum continuous rating (MCR) which is based upon engine performance at an air temperature 45°C, cooling water before charge air cooler 32°C, and air pressure 1.013 bar.

Ambient air temperature in the Canadian marine environment is normally lower than the air temperature at which the engine builders test data is based. Summer marine air temperatures would typically be 15 to 20°C while winter temperature is often -10 to -20°C in Arctic regions and 0 to -10°C in coastal areas. Due to the decrease in air temperature and the attendant increase of air density, the charge air manifold pressure of a turbo charged engine will increase due to the increase in mass flow. The charge air manifold pressure of a normally aspirated engine remains unaffected by ambient air conditions. The effects of changes of ambient air temperature on engine specific exhaust gas flow are shown by Figure 5.15 [52].

There are two types of heat recovery boiler systems which are used at sea; the wholly liquid phase and liquid/gaseous phase systems (hot water boilers or steam boilers). In recent years fluids other than water have been proposed and used, but are not common place.

Wholly liquid phase systems are very suitable if only heating loads are required and heat exchangers for this system tend to be less expensive than for liquid/gaseous phases boilers. Water as a heating medium has the advantage of low cost and availability and has a higher specific heat than other fluids. The use of water is limited by its relatively low boiling point. If a pressurized system is adopted to increase the temperature then much of the advantage of a fluid system is lost due to a large increase in the cost of equipment.

The use of fluids, other than water, provide significant advantages for special applications. The principal advantage of the thermal fluid is that higher temperatures may be used although this advantage is somewhat offset by the lower specific heat of the fluid than water. Fluids are available which are capable of operating over an extended range of temperatures, have high thermal stability and conductivity coupled with low viscosity. The range of fluids presently available include [53, 54]:

- Parafinic and napthenic based mineral oils (the most commonly used fluids). Range -25°C to 315°C.
- Synthetic oils. Range -25°C to 350°C.
- Diphenyl oxide.
- Inorganic salts.
- Mercury.

The last two fluids are only used in specialised high temperature processes, the first three fluids are the most useful for shipboard systems. The lower operating temperatures given are the minimum at which a single stage centrifugal pump will handle the fluid corresponding to a viscosity of 300 cSt.

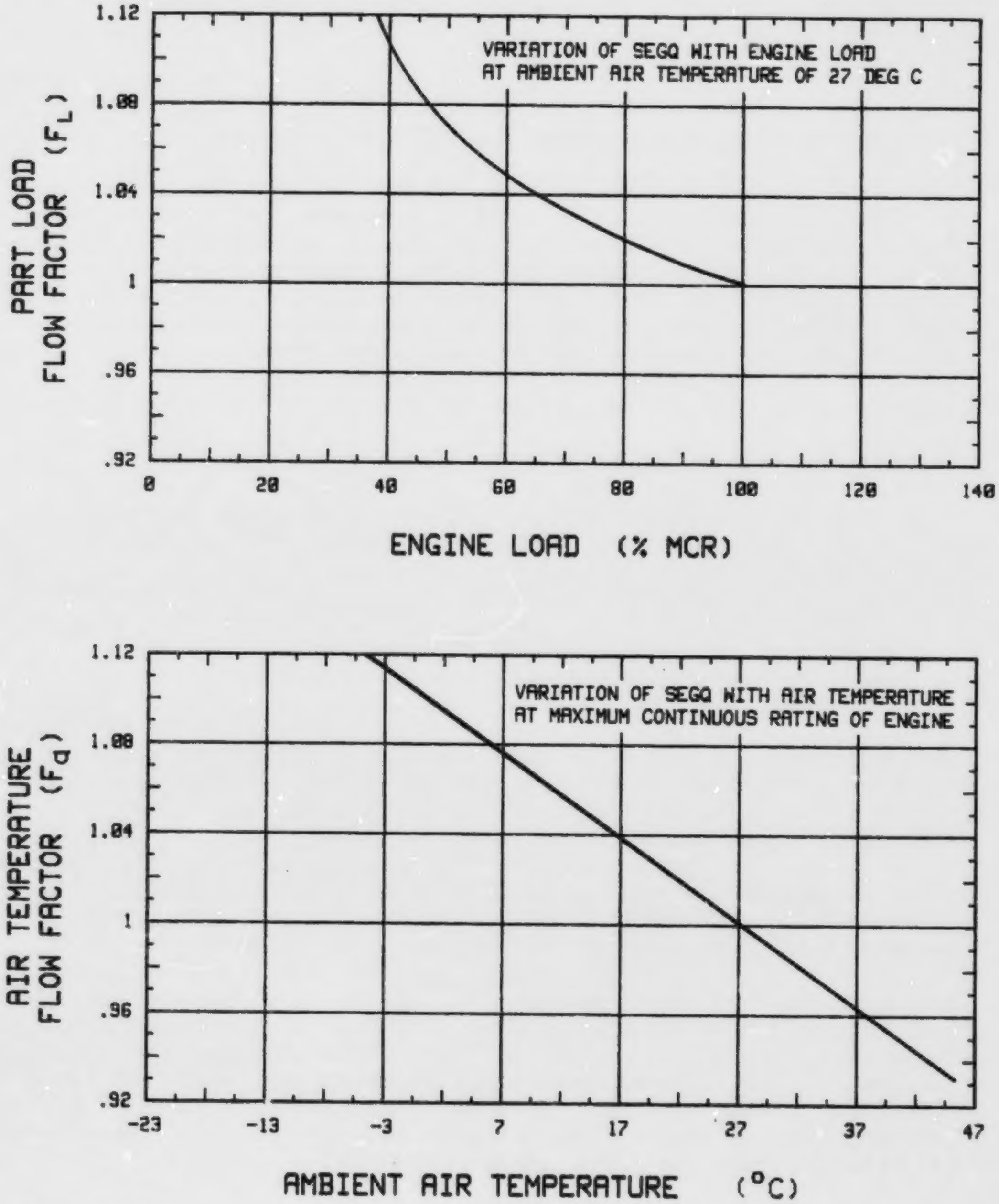
Although "thermal" fluids have been installed in more than 200 vessels [53] there is a clear reservation by the marine industry to adopt this more exotic form of heat distribution system unless a clear technical and commercial advantage can be demonstrated. In this respect the case for the use of these fluids in place of more conventional water and steam systems has yet to be made.

The use of gases with low latent heat of evaportion in preference to steam in the Rankine cycle (boiler/turbine system) can improve the efficiency significantly. Depending upon the choice of gas, a lower temperature heat source than is practical for steam systems can be used. Although this system provides indisputable advantages of efficiency over the traditional steam systems it has yet to be fitted to a ship, probably due to a higher capital cost and the cost of makeup gas in the event of leakage.

The basic exhaust gas waste heat recovery system uses a finned tube heat exchanger (economiser) in the exhaust gas path. A separately fired auxiliary boiler can be used as the steam receiver and water is circulated from the auxiliary boiler through the waste heat economiser and returned as

FIGURE 5.15

SPECIFIC EXHAUST GAS QUANTITIES (SEGQ)
VERSUS ENGINE LOAD AND AMBIENT TEMPERATURE



[50, 52]

NOTE: TYPICAL SPECIFIC EXHAUST GAS MASS FLOW = 8.0 kg/kWh
AT MCR AND ISO CONDITIONS

saturated steam to the boiler. If the evaporation rate of the waste heat economiser system is in excess of ship service steam and bunker heating requirements, the excess steam can be used to generate electrical power by means of a turbine driven generator. However, turbines which operate with saturated steam at the throttle are less efficient than equipment which used superheated steam and may give rise to problems of erosion due to wet steam in the last stages of the turbine. Figure 5.16 shows a simple basic single evaporation steam condensation system without a turbo-alternator.

If a turbine is to be driven from steam supplied from the waste heat boiler, the steam at the turbine throttle valves should be superheated. This is presently the most common system and is shown in Figure 5.17.

The efficiency of the waste heat boiler is a direct function of the difference between the exhaust gas temperature before the boiler and the gas temperature after the economiser section of the boiler, i.e., T_1-T_4, as shown in Figure 5.17. Temperature T_4 is determined by the dew point of the exhaust gases and is normally not less than 160°C. Therefore the feed water inlet temperature to the economiser must always be sufficiently low to achieve the maximum permissible temperature drop across the economiser section, i.e. T_4 should never be appreciably more than the dew point. Therefore feed heating has limited use in waste heat recovery systems.

Waste heat boiler design is based upon four basic parameters:

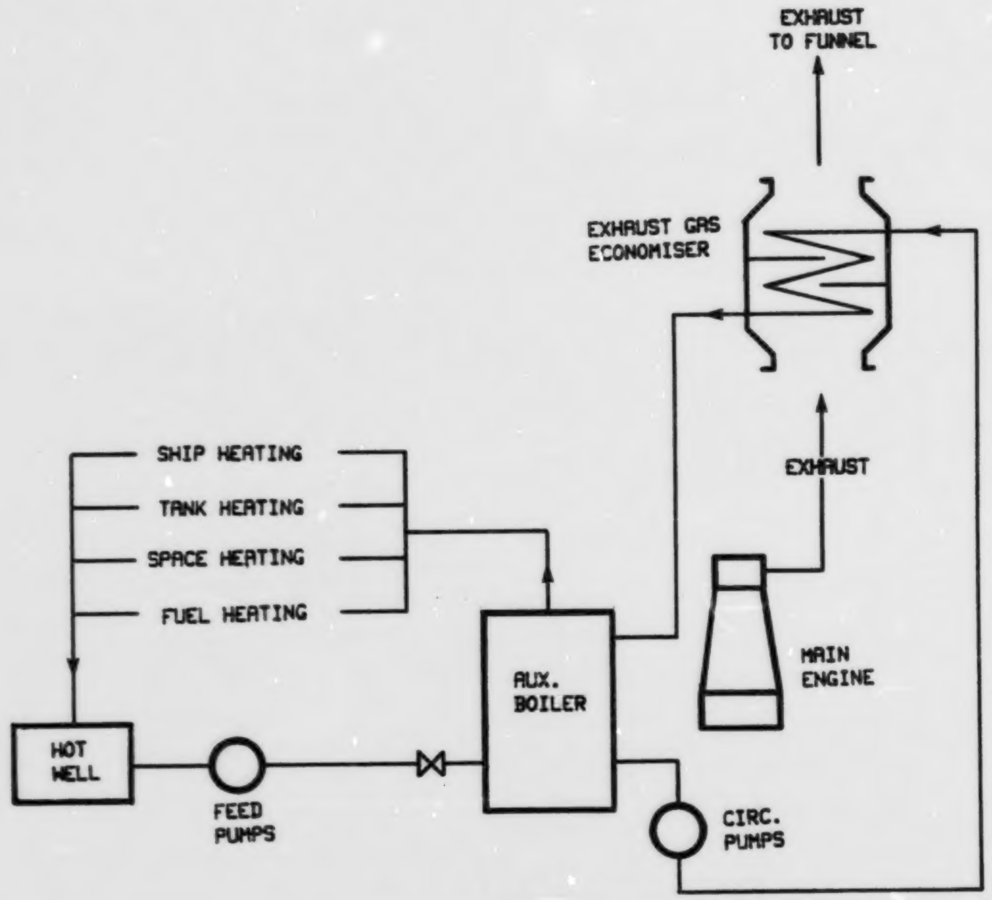
- Pinch point, defined as the difference between the gas temperature after the evaporation section of the boiler and the saturated steam temperature (10 - 25°C).
- Approach temperature, defined as the difference between the exhaust gas temperature before the superheater and the outlet steam temperature from the superheater (40°C).
- Specific heat of the exhaust gases (1.1 kJ/kg°C).
- Radiation loss (3%).

The dual pressure boiler is essentially two boilers in one casing. The first is a superheat boiler with stop valve steam conditions of about 10 to 15 bar at 285-300°C and the second boiler is a saturated steam boiler having stop valve conditions of 3 to 4 bar, saturated. The dual pressure boiler system is designed for use with a mixed pressure turbine. The high pressure steam is expanded through the turbine throttle valves and steam from the low pressure boiler is used for ship service and bunker heating, and the balance expanded through the low pressure stages of the turbine.

The mass flow of steam generated by the low pressure and high pressure boilers combined is slightly greater than that generated by the single drum boiler but the output from the turbine is significantly greater due to the increase in total thermal efficiency permitted by the use of higher steam conditions to the turbine. The upper steam pressure can be fixed at any convenient value to suit the turbine designer and will improve turbine efficiency through the higher steam conditions.

FIGURE 5.16

BASIC SINGLE PRESSURE STEAM-CONDENSATE SYSTEM FOR EXHAUST GAS HEAT RECOVERY


EXHAUST TO FUNNEL
EXHAUST GAS ECONOMISER
SHIP HEATING
TANK HEATING
SPACE HEATING
FUEL HEATING
EXHAUST
MAIN ENGINE
AUX. BOILER
HOT WELL
FEED PUMPS
CIRC. PUMPS

[50]

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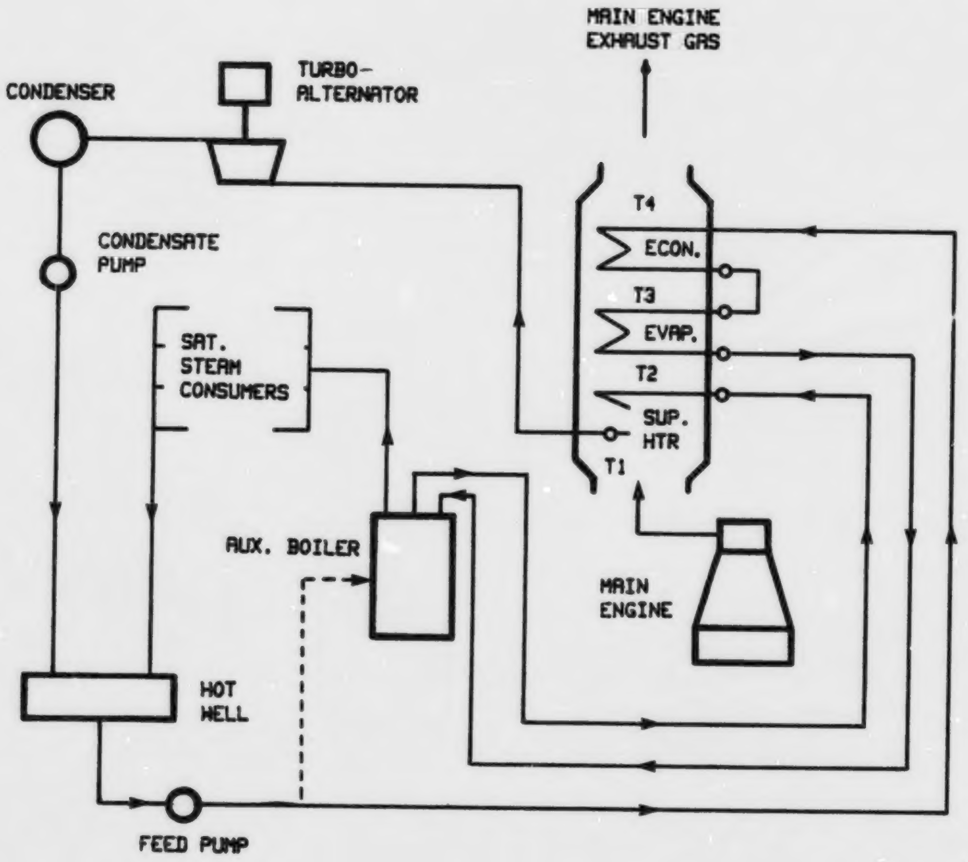
FIGURE 5.17

MODERN SINGLE PRESSURE STEAM-CONDENSATE SYSTEM FOR EXHAUST GAS HEAT RECOVERY


MAIN ENGINE
EXHAUST GAS
CONDENSER
TURBO-
ALTERNATOR
T4
ECON.
T3
EVAP.
T2
SUP.
HTR
T1
CONDENSATE
PUMP
SAT.
STEAM
CONSUMERS
AUX. BOILER
MAIN
ENGINE
HOT
WELL
FEED PUMP

[50]

MELVILLE SHIPPING.LTD

Figure 5.18 shows a simplified schematic diagram of a dual pressure steam system for exhaust gas recovery. For this application feed heating by the use of heat from the main engine jacket cooling water would be used. It can be shown that the percentage gain for a dual pressure system over the single pressure system is dependent upon the superheat boiler design pressure. The power from the turbo-alternator would be in the region 10 to 20% greater than a similar unit supplied with steam from a single evaporation boiler.

b) Recoverable Heat

Estimates have been made of the recoverable heat and electrical power generation potential for typical marine slow speed and medium speed diesel engines. The energy available, the energy recoverable and the turbo alternator output have been calculated with an engine exhaust temperature of 275°C and 354°C for a slow speed and medium speed engine respectively. The results are shown in Figures 5.19 and 5.20. Included in the calculations are estimates of 1,000 kg/hr of saturated steam or 650 kW thermal for ship's auxiliary heating requirement for bunker heating and other services. The waste heat boiler gas outlet temperature has been assumed to be 160°C. For part load conditions, the flow factors of Figure 5.15 should be used. Due to the higher exhaust gas temperature of medium speed engines the energy which can be recovered is considerably higher than from slow speed long stroke engines.

The Figures 5.19 and 5.20 show the overall loss of energy from the exhaust gases. The quantity of energy which is recoverable as shaft or turbo-alternator power is only a small proportion of the total energy in the gases. If all the steam generated in the exhaust gas boiler was required for heating purposes, the thermal efficiency of the cycle would be much improved as the latent heat of evaporation would be utilised. However the potential for steam generation from the engine waste heat usually is in excess of that required for ship heating purposes and is best utilised for electrical power generation or to augment the propulsion power, despite the inefficiency of the turbine engine.

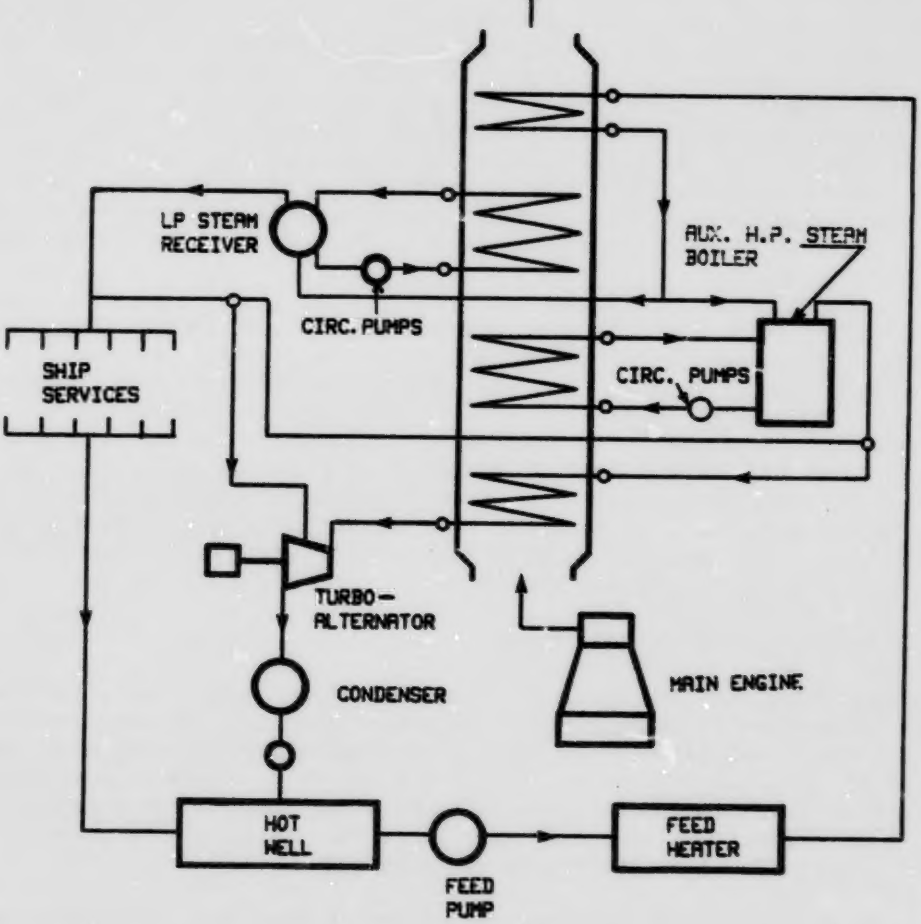
c) Generation of Electrical Power

The potential for the generation of electrical power by a turbine driven alternator from steam supplied by a single pressure and dual pressure waste heat recovery boilers fitted to a slow speed and medium speed diesel engine is shown on Figures 5.21 and 5.22. The single pressure boiler calculations have been based upon no condensate heating and the dual pressure system with condensate heating. All technical data for calculations have been taken from catalogues of diesel engine manufacturers.

Figures 5.21 and 5.22 show the power output from the turbo-alternator as a percentage of main engine power at MCR rating. Electrical output for engine powers other than MCR must be separately calculated by the use of part load factors. If no steam is used for ship heating then the medium speed installation can provide approximately 8-10% of main engine power from exhaust gas waste heat recovery and the slow speed long stroke installation 4-6%. The reduction of output from the turbo-alternator system is shown after 1,000 kg/hr of saturated steam have been drawn from the boiler for ship heating needs.

FIGURE 5.18

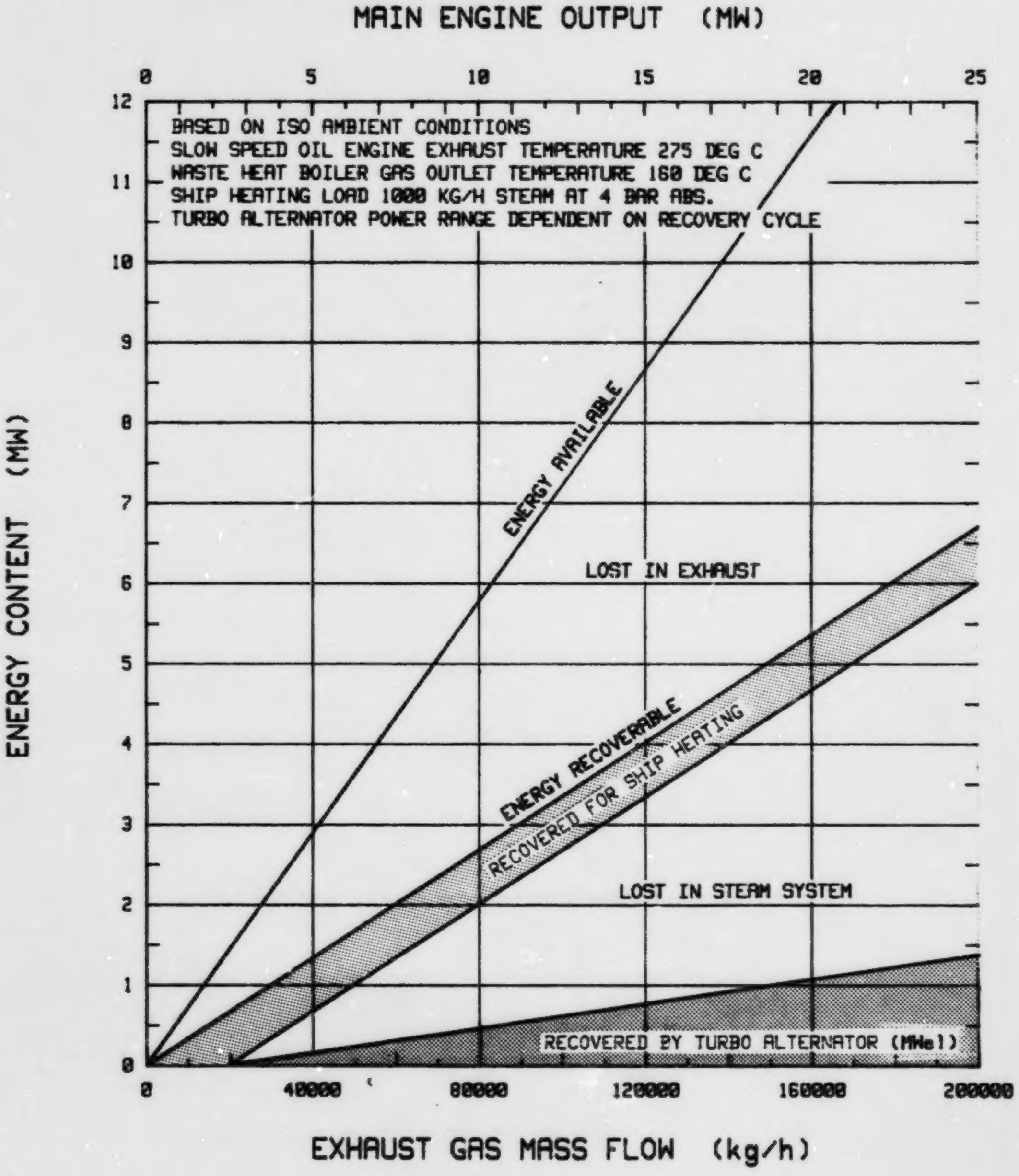
MODERN DUAL PRESSURE STEAM-CONDENSATE SYSTEM FOR EXHAUST GAS HEAT RECOVERY


LP STEAM RECEIVER
CIRC. PUMPS
AUX. H.P. STEAM BOILER
SHIP SERVICES
CIRC. PUMPS
TURBO-ALTERNATOR
CONDENSER
MAIN ENGINE
HOT WELL
FEED PUMP
FEED HEATER

[55]

FIGURE 5.19

SLOW SPEED OIL ENGINE WASTE HEAT RECOVERY FROM EXHAUST GAS

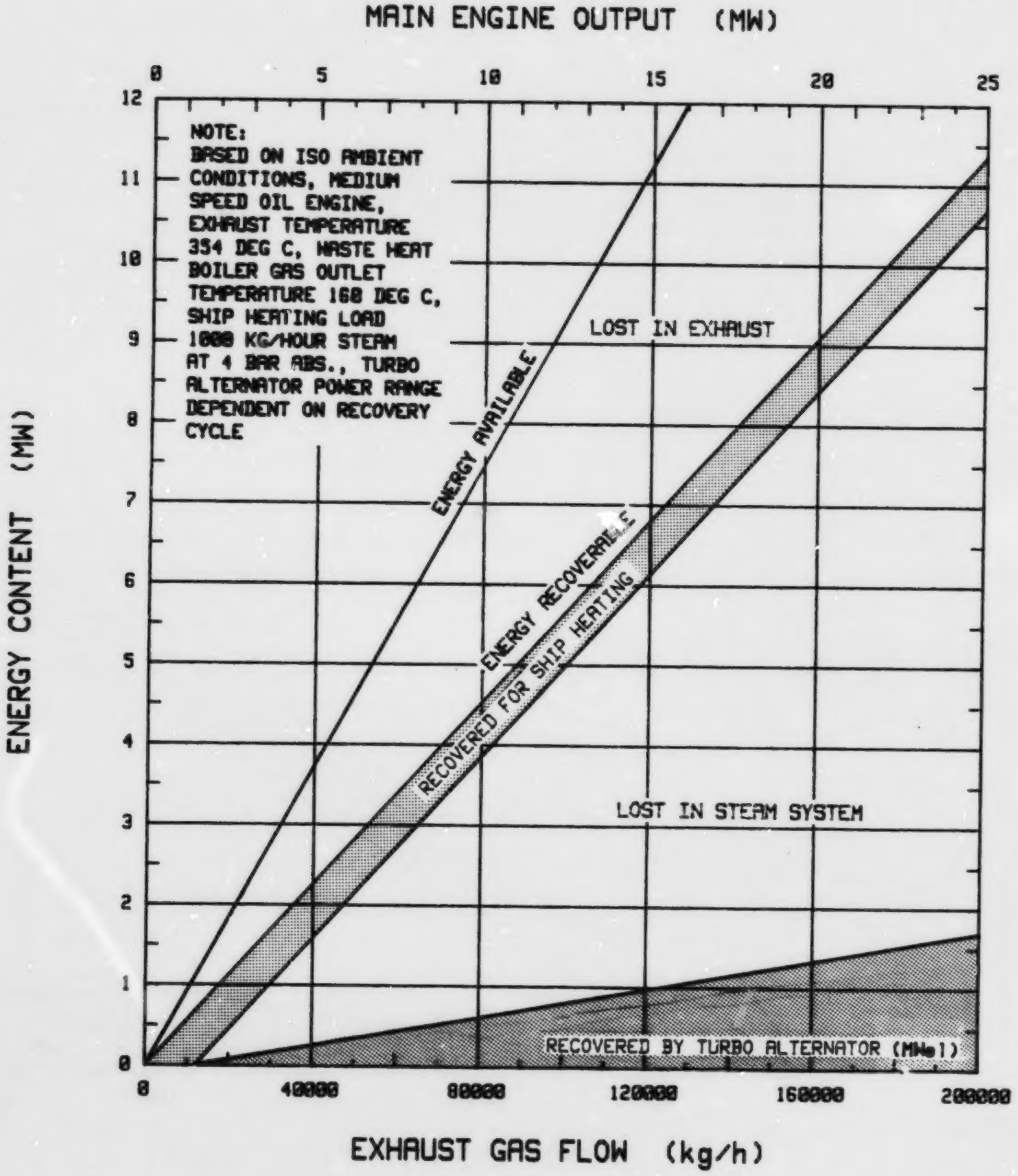


BASED ON ISO AMBIENT CONDITIONS
SLOW SPEED OIL ENGINE EXHAUST TEMPERATURE 275 DEG C
WASTE HEAT BOILER GAS OUTLET TEMPERATURE 160 DEG C
SHIP HEATING LOAD 1000 KG/H STEAM AT 4 BAR ABS.
TURBO ALTERNATOR POWER RANGE DEPENDENT ON RECOVERY CYCLE

* SPECIFIC EXHAUST GAS QUANTITY 8 kg/kWh

FIGURE 5.20

MEDIUM SPEED OIL ENGINE WASTE HEAT RECOVERY FROM EXHAUST GAS

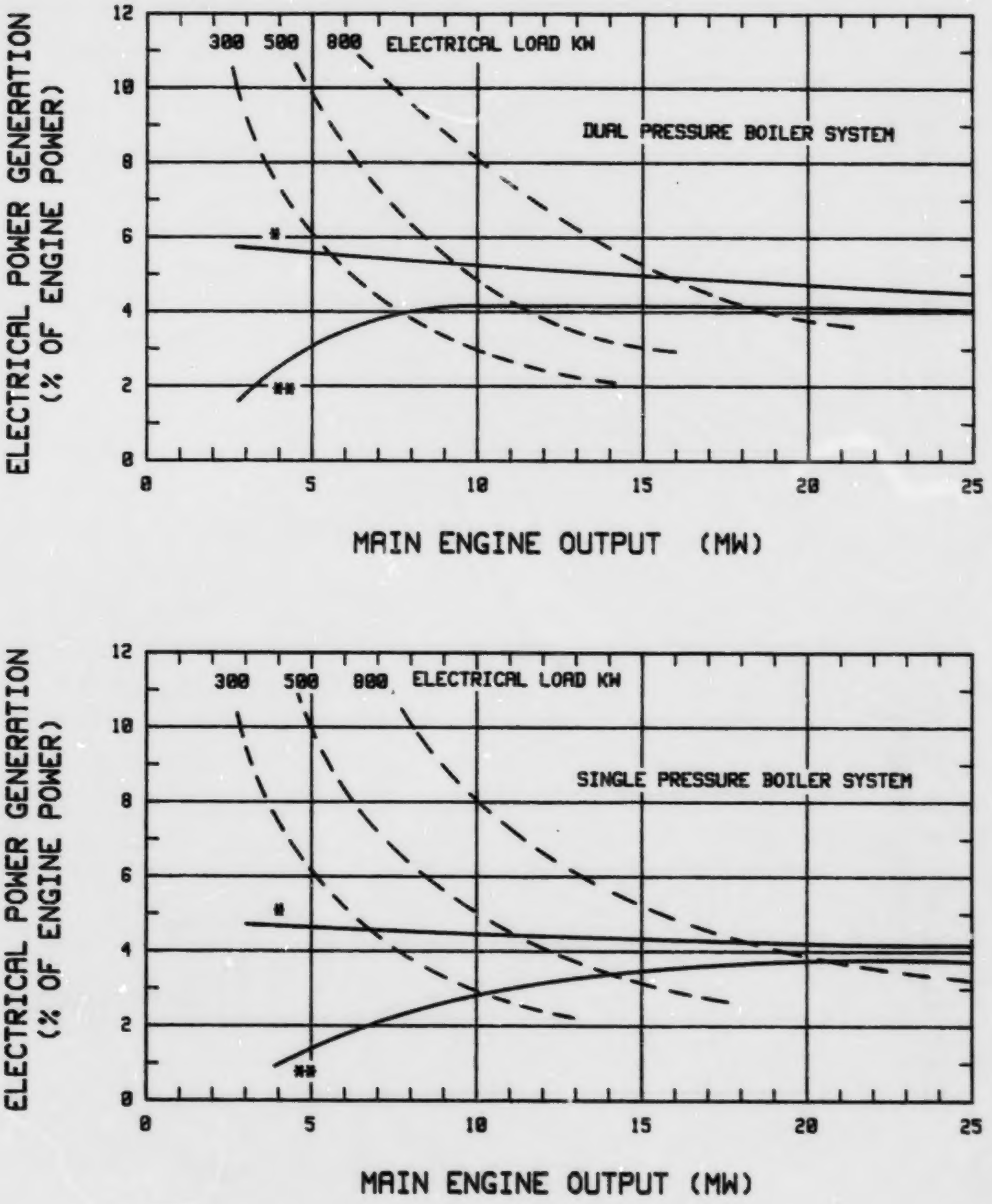


NOTE:
BASED ON ISO AMBIENT CONDITIONS, MEDIUM SPEED OIL ENGINE, EXHAUST TEMPERATURE 354 DEG C, WASTE HEAT BOILER GAS OUTLET TEMPERATURE 160 DEG C, SHIP HEATING LOAD 1000 KG/HOUR STEAM AT 4 BAR ABS., TURBO ALTERNATOR POWER RANGE DEPENDENT ON RECOVERY CYCLE

* SPECIFIC EXHAUST GAS QUANTITY 8 kg/kWh

FIGURE 5.21

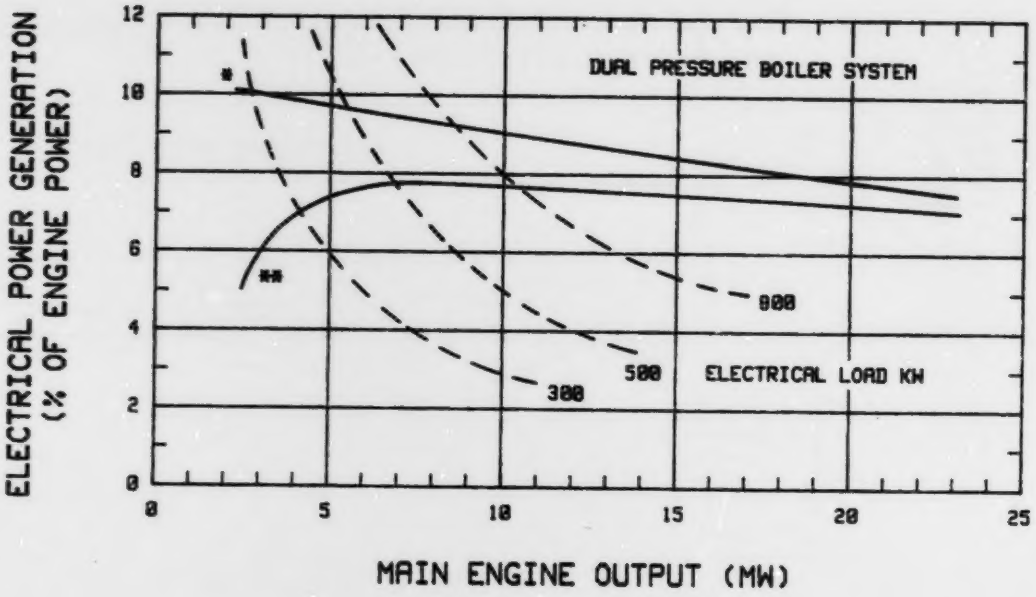
ELECTRIC POWER GENERATION AS A FUNCTION OF ENGINE POWER FOR SLOW SPEED OIL ENGINES

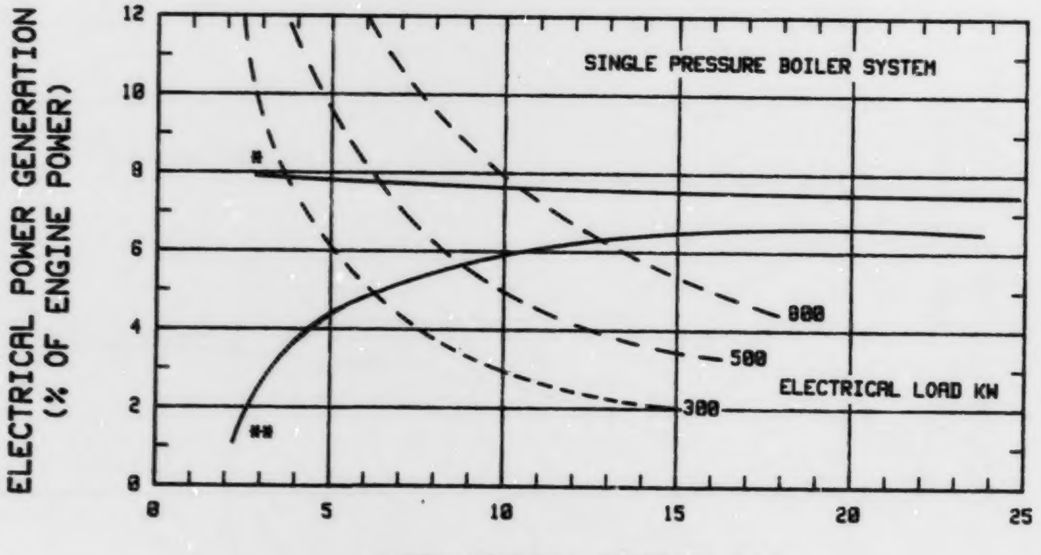


* No ship heating steam
** 1000kg/hr saturated steam to ship heating load

FIGURE 5.22

ELECTRIC POWER GENERATION AS A FUNCTION OF ENGINE POWER FOR MEDIUM SPEED OIL ENGINES





* No ship heating steam
** 1000 kg/hr saturated steam to ship heating load

The electrical power requirement while at sea will differ substantially between different types of vessels. Electrical load of 300, 500 and 800 kilowatts have been shown on Figure 5.22. The electrical load that can be supplied from the single and dual pressure boiler systems fitted to a two stroke slow speed engine and a four stroke medium speed engine can be deduced from the figure.

Table 5.5 has been derived from Figures 5.21 and 5.22 and shows considerable variations for the two types of engine and the type of steam system. For example, with an electrical sea load of 500 kW a simple single pressure waste heat recovery system would require slow speed engine power of about 14,000 kW whereas a medium speed engine with an integrated dual pressure steam cycle would only require about 7,000 kW. Such large variations must be assessed during the design stage of the propulsion plant and auxiliary system and once again the type of ship and its trading pattern will have a heavy influence on the system to be fitted.

It is a common belief that unless the main engine can support the total electrical sea load at normal operating conditions then the use of a turbo-alternator heat recovery system would be uneconomical. This is incorrect, and every ship should be rigorously evaluated. To achieve the maximum efficiency the aim must be to utilise as much of the fuel energy as possible, requiring the most efficient waste heat recovery system to be installed. The provision of such systems requires additional capital investment and the commercial balance between capital and operating costs must be achieved for total economic efficiency.

Utilisation of exhaust gas supported turbine power, although easily adapted to electrical power production, may also be used to supplement main propulsive power. This can be done either by the turbo-alternator or by an electrical motor/generator, in both cases connected to the reduction gear. The motor/generator design needs two electrical machines, one combined with steam turbine and the other geared to propeller shaft. The surplus of electric power from a turbo-genertor is fed back to the propulsion system by means of electrical connection.

In the case of turbo-generators geared mechanically to the propeller shaft a special gear assembly must be employed to maintain constant generator frequency and to enable power transmission in two directions. Such a system, designed by the Japanese IHI Co., and called Super Economical Shaft Generator System (SSGs) Mark I and II, has already been fitted to newly built ships in Japan [55].

5.4.3 Charge Air Cooling Water

The combustion air for modern engines is supplied by a turbo-charger, driven by exhaust gases from the engine. Figure 5.23 shows a charge air heat recovery schematic diagram and the principal uses to which the recovered heat may be put [49]. The turbo-charger compresses the air and the heat of compression is removed from the air before it enters the cylinder. This allows a larger weight of air to be admitted to the cylinder, more fuel to be burnt and more power to be developed. With the modern turbo-charger designs, considerably more air is supplied to the engine than is required for stoichiometric combustion. Depending upon engine and turbo-charger

TABLE 5.5

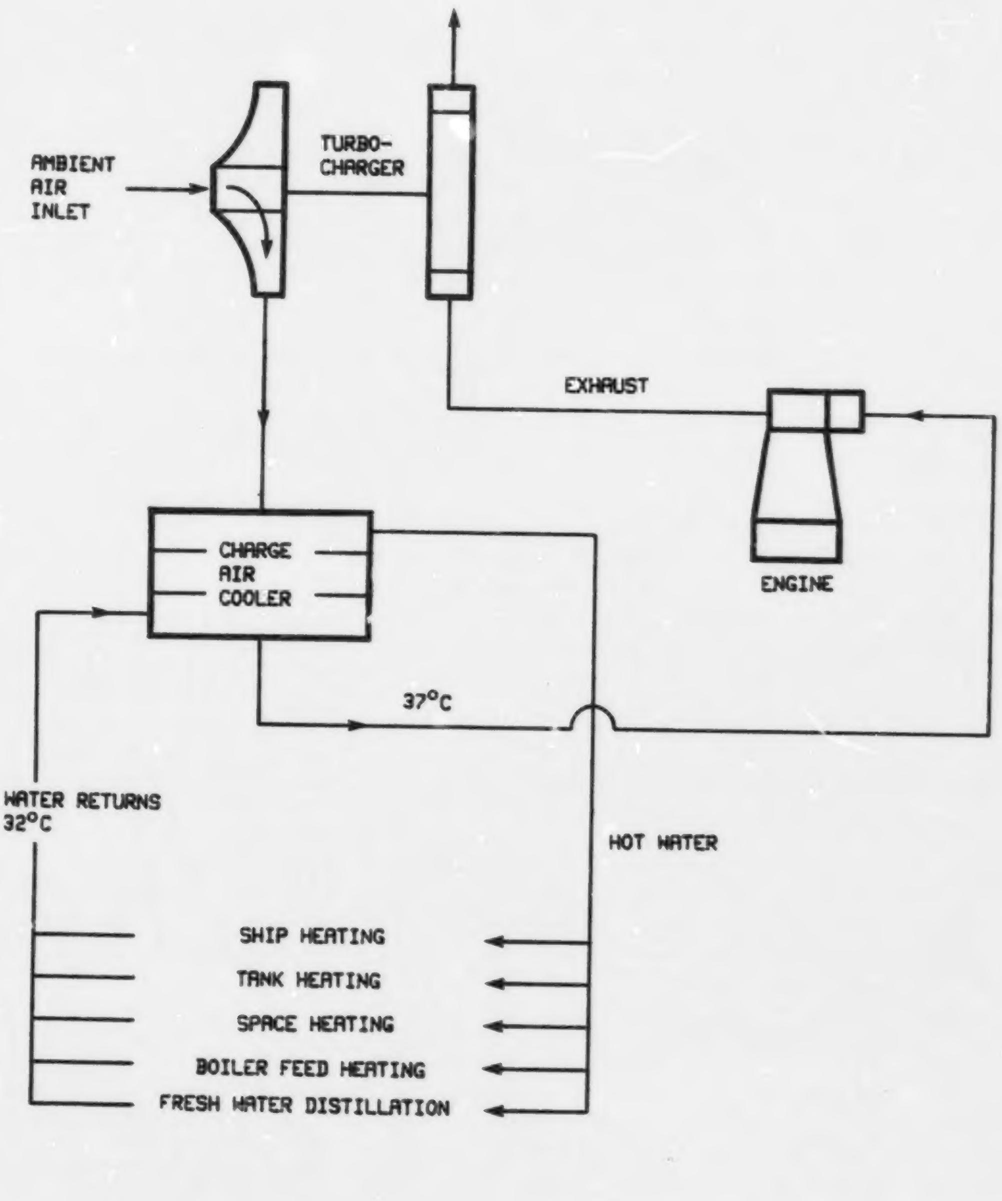
MAIN ENGINE POWER (kW) TO SUPPORT ELECTRICAL LOAD AT SEA

Main Engine Type	Exhaust Gas Heat Recovery System	Main Engine Power (kW) to Support Electrical Sea Load of: 300 kW	500 kW	800 kW
Two Stroke Slow Speed Long Stroke (No Turbo-Charger Gas Bypass)	Single Pressure	10,000	14,250	21,500
	Dual Pressure	7,800	11,750	17,500
Four Stroke Medium Speed	Single Pressure	6,000	9,000	12,800
	Dual Pressure	4,400	6,800	10,500

Note: Refer to Figures 5.27 and 5.28

FIGURE 5.23

CHARGE AIR HEAT RECOVERY SCHEMATIC
FOR TYPICAL MARINE ENGINES


AMBIENT AIR INLET
TURBO-CHARGER
EXHAUST
ENGINE
CHARGE AIR COOLER
37°C
WATER RETURNS 32°C
HOT WATER
SHIP HEATING
TANK HEATING
SPACE HEATING
BOILER FEED HEATING
FRESH WATER DISTILLATION

[49]

design, 50 to 150% in excess of that theoretically required for combustion is supplied to the cylinders, the additional air being used to purge the cylinders of combustion gases and to cool the cylinder liners, pistons, and valves. Historically the charge air was cooled by sea water and the heat dissipated into the sea. Present practice is to circulate the air coolers with fresh water and recover a proportion of the heat.

As engine load decreases so the quantity of exhaust gas correspondingly decreases and the pressure ratio across the turbo-charger air side and the temperature and pressure of the charge air after the compressor decreases. This may be seen in Figure 5.24. The temperature of the charge air after the compressor is dependent on the air inlet temperature to the compressor. This is also shown in Figure 5.24. It can be seen that for an engine operating at 85% of its maximum continuous rating with an air inlet temperature of 27°C the temperature after the turbo compressor would be approximately 140°C. Even with an air inlet temperature of 0°C the compressed air would reach a temperature of about 104°C at the same engine load. The figure illustrates that the heat recoverable from the charge air is dependent on the engine load and the air inlet temperature to the charge air compressor.

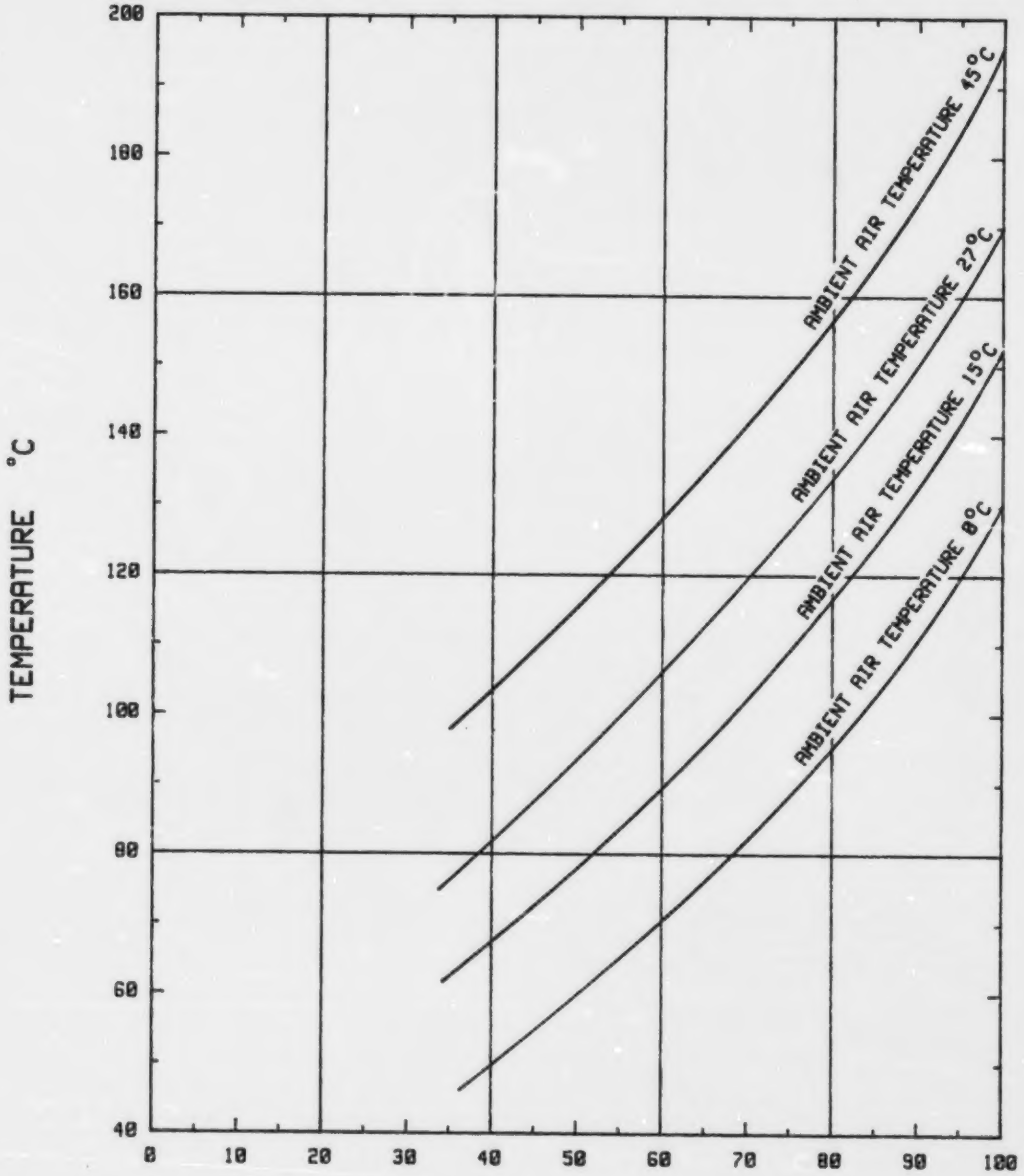
The heat recoverable from the charge air depends on the mass flow of air and temperature. Although the total air mass flow through the engine decreases with load the specific flow of air (kg air/kW hr) increases with decrease of load.

The combination of air temperature and mass flow provide the total heat available and Figure 5.25 shows this parameter for variations of engine load for various ambient air temperatures. From the figure it may be seen that the heat in the charge air is more dependent upon inlet air temperature than mass flow. As an example 21% of the engine output is available for recovery as heat from the charge air when the engine is operating at 80% MCR with an ambient air temperature of 27°C. Although this quantity of heat is a large proportion of the engine shaft output, the temperature is low and the use to which the recovered heat can be put is limited.

In a straight counterflow heat exchange system, typically used as a charge air cooler, the minimum temperature differential between the cooling water and the air to be cooled will not be less than 5°C at the water discharge from the cooler. If the cooling water is to be used for "all purpose" ship and bunker heating, then the cooling water temperature ex-cooler is required to be not less than 80°C and a temperature close to 95°C in a non pressurized system would be desirable. There will be certain conditions of engine load or ambient temperature when the necessary temperature differential will not be achieved. The chain dotted lines in Figure 5.25 show the limiting conditions with water outlet temperatures from the heat exchanger of 80, 95 and 110°C. As an example, water temperatures greater than 95°C cannot be achieved below engine loads of 55% MCR when the ambient temperature is 27°C. While this must be recognised at the design stage it is not serious as normally the vessel would operate at 80% MCR and above. Even at 0°C ambient air temperature 95°C outlet temperature from the charge

FIGURE 5.24

CHARGE AIR TEMPERATURE BEFORE COOLERS



[49]

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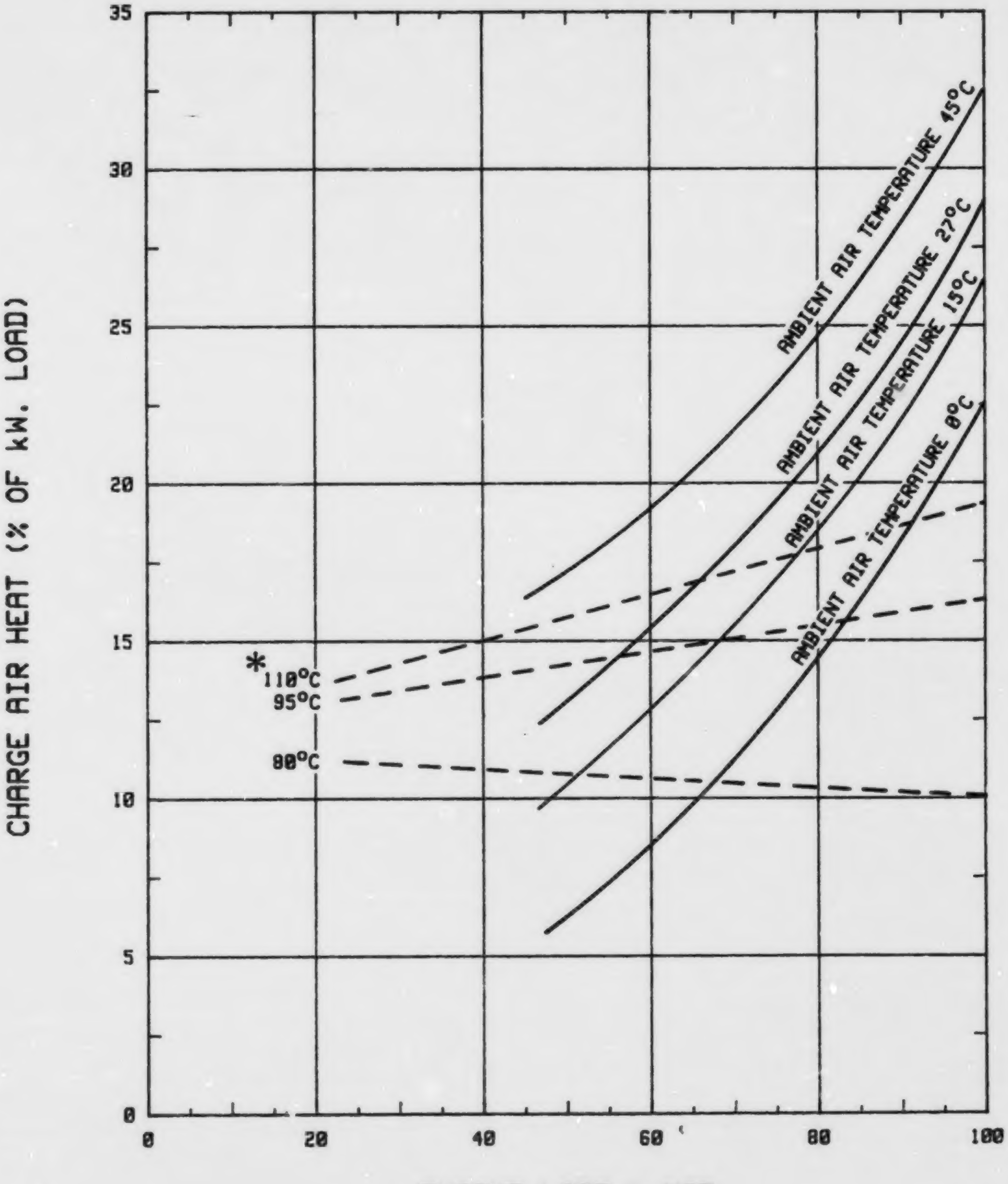
FIGURE 5.25

POTENTIAL HEAT QUANTITIES IN CHARGE AIR OF TYPICAL MARINE OIL ENGINES



[49, 50]

* WATER OUTLET TEMPERATURE FROM CHARGE AIR COOLER

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air coolers can be obtained. If the available heat is not required for ships heating or fuel heating purposes, it could be used to heat boiler feed water or other low temperature heat users such as bunker storage tanks.

5.4.4 Jacket, Turbo-Charger Casing and Piston Cooling Water

The other sources of low grade heat available from the diesel engine are the jacket, turbo-charger casing and piston cooling water but the temperature of the coolant from these sources is usually lower than that associated with the charge air. The system diagramtic for a jacket cooling water heat recovery is shown in Figure 5.26 [51]. Jacket water temperature is in the region 70-90°C and although quantity of heat available is of similar magnitude to the charge air, the uses to which the jacket water heat may be put are reduced by the water temperature. The jacket cooling water heat is often used in single or multi-stage fresh water evaporators. The heat could be used for ship's accommodation heating purposes or for boiler feed heating but the energy to be saved, as in all cases, must be weighed against the cost and complexity of the extra equipment required.

At present, most marine engines are fitted with water cooled turbo chargers, usually in the same water circuit as the engine cylinder jackets. In an effort to increase the overall efficiency of the engine system, engines are now marketed with uncooled turbo-chargers, increasing the quantity of heat recoverable from the exhaust gases. As shown in Figure 5.27, the effect of this design change is that the engine cooling dissipates less heat and the specific heat available for recovery from the cooling water decreases with engine load.

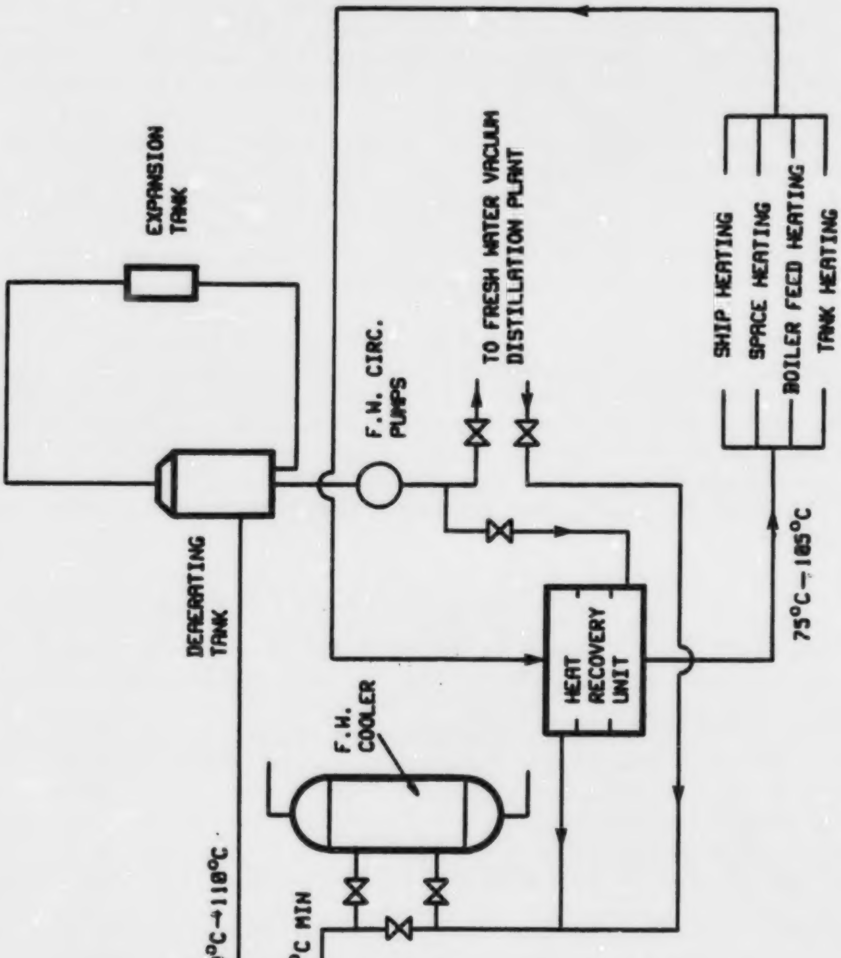
5.5 Pumps

The auxiliary power demand to support the the main engine is often 4-5% of the main diesel engine MCR and up to 10% of the power of a steam turbine vessel. The power is required to drive pumps and fans which in many cases are single speed machines electrically driven. In the case of a steam vessel the main feed water pumps and forced draft fans are in many cases turbine driven. The pump and pump prime movers are fitted to satisfy the engine MCR requirement for all service conditions and hence the pump capacity is often greatly in excess of that required for normal continuous operation, operation in temperate climates and at part load operation.

Generally, constant speed electrical induction motors are used to power pump and the pump output is not controlled. This often results in uneconomical operation due to the pumping of large quantities of cooling water which is superfluous to requirements. Not only are these energy losses due to inefficient operation of pumps, the pump prime movers are also generally inefficient. Most pumps onboard ship are driven by constant speed induction motors which are generally 92-96% efficient depending upon size, the generators which supply electrical energy to the pumps are about 35% efficient, if diesel driven, and no more than 20% if steam turbine powered. If the pumps are driven by a steam turbine, the thermal efficiency of the turbine is often less than 15%. Hence, auxiliary power efficiency is at best about 35% efficient and at worst less than 10% efficient.

FIGURE 5.26

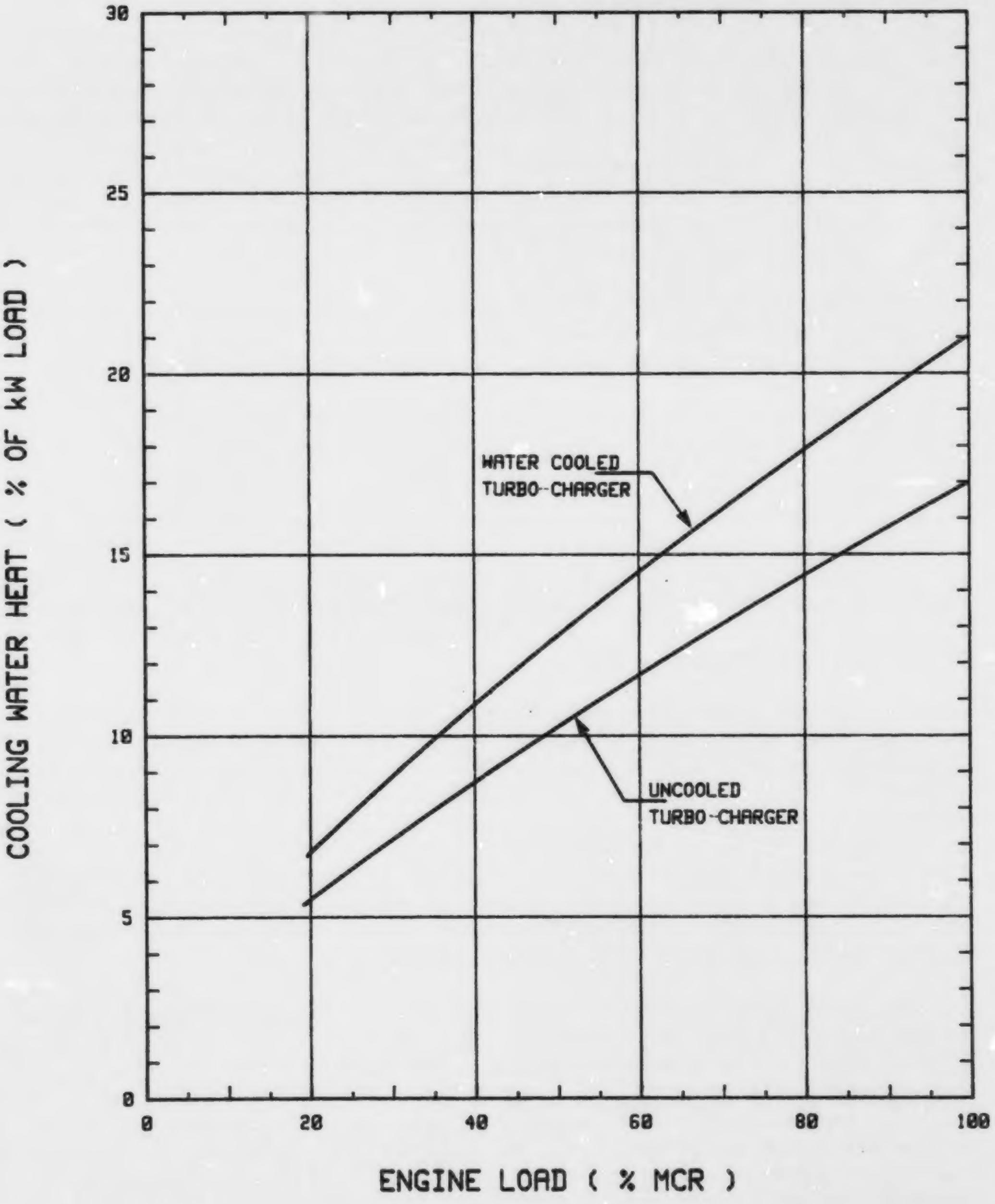
JACKET COOLING WATER HEAT RECOVERY SCHEMATIC FOR MARINE OIL ENGINES


EXPANSION TANK
DEAERATING TANK
F.W. CIRC. PUMPS
TO FRESH WATER VACUUM DISTILLATION PLANT
SHIP HEATING
SPACE HEATING
BOILER FEED HEATING
TANK HEATING
75°C→105°C
HEAT RECOVERY UNIT
F.W. COOLER
80°C→110°C
45°C MIN
MAIN ENGINE

[51]

FIGURE 5.27

TYPICAL HEAT QUANTITIES IN JACKET COOLING WATER OF MARINE OIL ENGINES



[51]

Much attention has been given in recent times to reducing the pumping load by the use of:

- variable speed motors
- main engine driven pumps

5.5.1 Variable Speed Pumps

For vessels designed for unrestricted trading, heat exchanger and pump cooling systems are usually designed to permit maximum continuous operation of the main machinery under tropical conditions, i.e., 45°C air temperature, 32°C sea water temperature. However the majority of vessels spend much of their time in climates where the sea water temperature is well below 30°C and the cooling pumps need not deliver maximum capacity constantly. Similarly the flow of cooling water and hence pumping power can be reduced when the main engine is partly loaded.

If the quantity of cooling water or air can be regulated according to the sea water, air temperature and engine load requirement, a significant saving of auxiliary power is achievable. Various methods have been proposed:

- Two speed electric motors with throttling at low speed.
- The "running" and "standby" pump of a two pump system. A two speed motor can be fitted to one pump.
- Stepless variable speed using an eddy current coupling.
- Stepless variable speed using a variable frequency controller with a constant speed motor.

Analysis showes that the two speed motor method would give a payback time of 1.2 to 2.6 years. The two pump system is more expensive but the second pump need not be of the same rating.

Control of the pump speed by use of an eddy current coupling is achieved by varying the slip value between motor and pump coupling by means of regulation of the excitation current of the coupling. This system with automatic control circuits has performed well under rigorous testing on a test plant of Nippon Kokan KK and onboard a 18,000 dwt cargo vessel owned by Mitsui O.S.K. Lines. The calculations made by NKK and based on an 18PC 2-5V Pielstick diesel engine revealed that at 85% MCR and sea water temperature 20°C the application of this system for main sea water pumps saved 0.3% of the main engine MCR. The manufacturer claims that the extra cost of this system can be recovered within approximately one year.

Variable frequency control of pump motors is the most efficient system. The frequency and hence the speed of the squirrel cage motor is adjustable within wide limits providing smooth speed control from zero to above normal full speed. The present day cost of a pump unit with frequency inverter will be between 1.3 - 2.0 times the cost of the conventional unit. This higher cost can be offset by lower maintenance costs in addition to power savings. Another advantage of this system is the possible elimination of control valves in the cooling system by direct speed variation which will often simplify automation. Variable speed pumps have found particular application in tanker operation for cargo pumping systems [57, 59].

5.5.2 Main Engine Driven Pumps

Some diesel engine manufacturers have investigated the possibility of driving main engine pumps directly from the engine. Pielstick, for example, intends to build an engine front end drive into a gearbox with single input and multiple output. A single electric motor drive can also be provided into the pump power take off gear box. A similar project using this system was presented recently by J.E. Church. This concept is shown in Figure 5.29 [58, 59, 60].

The system shown by Figure 5.28 ensures automatic transfer of drive from main engine to electric motor due to sudden slow down or stop, whilst maneouvring, or when warming the engine. It is accomplished by a system of three speed increasing automatic self-disconnecting gearboxes. The increasing speed transmission also allows the use of standard high speed pumps at no extra cost.

The main engine driven pumps make the main engine self sufficient and give the best economic results in conjunction with a waste heat boiler and a steam turbo driven generater. This enables all auxiliary diesel engines to be shut down at sea ensuring substantial fuel savings.

A similar approach has been proposed by use of power take-offs from the gearbox of medium speed engine installations.

5.6 Summary

All auxiliary systems of a vessel, if carefully designed and operated, offer remarkable savings in fuel consumption. Usually the greatest potential for savings is in auxiliary systems producing electrical power.

Conventional arrangements of diesel engine or steam turbine driven generators are more efficient today due to improvements in prime movers. The largest savings may be achieved by use of the main engine, which is usually the most efficient prime mover onboard ship, to supply all electric power requirements. This involves the use of either; waste heat to power the electrical generating systems, or main engine driven electrical generators.

Power take-off from the main engine is very popular in propulsion plants with medium speed diesel engines geared to a CP propeller since alternators are constant speed machines and ships with CP propellers could easily operate at constant RPM regardless of load. This is not possible in slow speed diesel engines with fixed pitch propellers unless the alternator is provided with frequency converters, usually at high cost. Moreover, the physical size and therefore cost of a generator fitted to a slow speed engine is excessive unless gearing is incorporated.

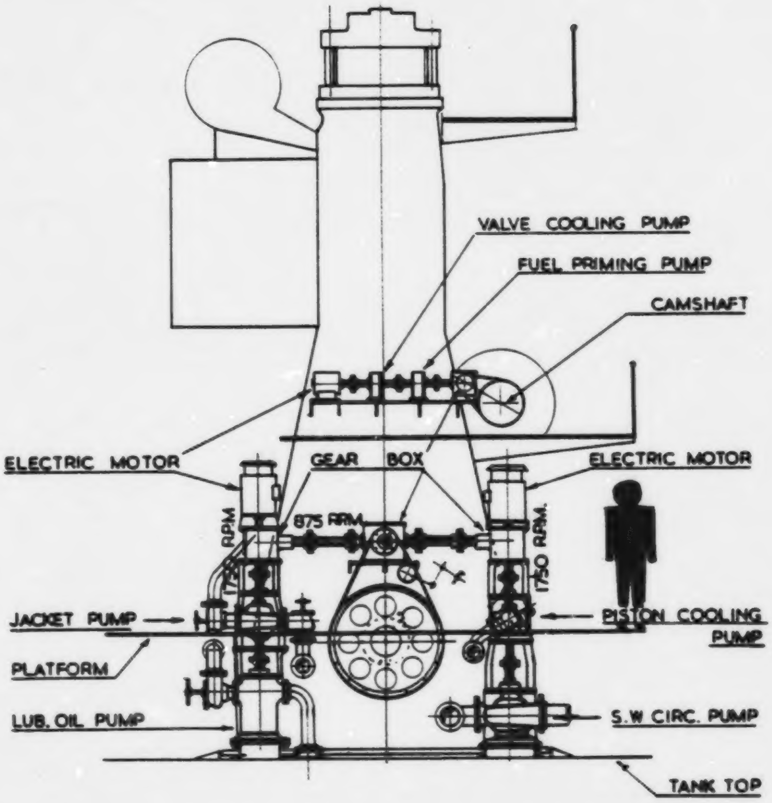
The other source of savings comes from minimizing heat losses from the machinery by the use of waste heat recovery systems. Heat is usually recovered from the exhaust gases, and the charge air and cooling water.

Exhaust gases are utilized more often, for generating electric power especially on larger vessels, by means of steam turbo generators; the other heat sources serves mainly in various heating arrangements. Through introducing newly developed components of waste recovery systems, such as dual pressure steam boilers, the overall efficiency of heat recovery has increased significantly and new shipowners can expect larger savings relative to the investment made in the necessary equipment. For a relatively small investment, quite large savings can be achieved, but as the percentage of heat to be saved increases, the investment in equipment also increases considerably.

Waste heat recovery is very attractive to shipowners since, once the initial investment has been made, other than maintenance costs, the benefits are at no cost.

FIGURE 5.28

DRIVE ARRANGEMENT OF MAIN ENGINE DRIVEN PUMPS


VALVE COOLING PUMP
FUEL PRIMING PUMP
CAMSHAFT
ELECTRIC MOTOR
GEAR BOX
ELECTRIC MOTOR
875 RPM
1750 RPM
1750 RPM.
JACKET PUMP
PISTON COOLING PUMP
PLATFORM
LUB. OIL PUMP
S.W CIRC. PUMP
TANK TOP

6.0 SHIP RESISTANCE

6.1 General

The total ship resistance, R_{TS} is generally given in the form of:

$$R_{TS} = R_{FS} + R_{RS}$$

where:

R_{FS} = frictional resistance
R_{RS} = residual resistance

Frictional resistance is due mainly to viscous friction created by the relative motion of the hull and the surrounding water. Residual resistance is the wave and wake components. Therefore, the reduction of the overall resistance of the ship, can be achieved by making a reduction in either or both of these resistance components.

In the following discussion, several methods of achieving resistance reductions will be examined.

6.2 Bow and Stern Bulbs

As a ship proceeds through the water, two wave systems are created, one by the bow and the other by the stern, absorbing energy from the ship's propulsive system. The principle of the bulbous bow is based on the reduction of this wasteful dissipation of energy.

In 1928, Havelock calculated the surface wave pattern due to a sphere immersed in a uniform stream [61]. The most important feature of this pattern was a wave trough just after the sphere, which suggested the possibility of partially cancelling the bow wave of the hull by locating a sphere below the stern. Wigley carried out the calculations of wave profiles and wave making resistance based on Havelock's work, and published the basic theory for the bulbous bow in 1935 [62]. At low speeds, Wigley found the total resistance to increase owing to the additional friction and form drag of the bulb. At high speeds, the reduction in wave resistance due to interference between the wave systems of the hull and bulb is more than sufficient to overcome the frictional and form drag of the bulb, and the net result is a reduction in total resistance.

Professor Inui in Japan has devoted much effort to the design of bulbs, with applications at both stem and stern, to annul the wave systems normally arising at those points [63]. In comparison with the stern bulb, the bow bulb began its application much earlier and was at one time the most common bow form for VLCC's and bulk carriers.

Given in Table 6.1 is a comparison of powering requirements for a 150 foot long trawler hull derived from model tests [64]. This comparison gives the derived power requirement for a range of hulls from the parent hull without the bulb through to that with a maximum bulb size of 15% of maximum ship section area. It can be seen that compared with the parent hull, the difference in power generally changes with speed. It can be observed that the bulb caused an increase in required power for speeds below 11.5 knots. For higher speeds, power saving was achieved to a maximum of approximately 10% at 13.5 knots.

It should be noted that, although the "break even" speeds observed in this comparison is not particularly high, the length of the hull is also small. Since equal Froude numbers are required for dynamic similarities, it follows then that for a geometrically similar ship of say 900 ft. long, the break even speed would have to be approximately 2.5 times higher to achieve the same Froude number. That is, for a ship 900 ft. long, the "break even" speed will be approximately 28 knots. Therefore, large bulbs would probably not be suitable for large ships and smaller bulbs must be chosen instead. It is due to this break even speed that in recent years with the trend of decreasing ship speeds, bulbous bows are gradually disappearing from applications in ships such as large tankers and bulk carriers.

Figure 6.1 shows the arrangement of a range of stern bulbs developed by Professor Inui, as applied to a container carrying vessel. Fuel reduction for this application was reported to be between 6 and 7.2%, an estimated 1,800 tons of fuel per year for a 27,000 kW container ship.

6.3 Frictional Resistance Control

Skin friction is believed to account for up to 80% of the total resistance for slow steaming tankers and bulk carriers [65], and hull roughness which has a direct bearing on friction can mean a large difference in fuel costs.

To illustrate the point, it has been reported that a typical 250,000 DWT tanker with its hull freshly coated and smooth, will run at 13 knots with an engine output of approximately 11,000 kW. A year later, 12,000 kW will be required; at the end of two years, its engine must develop 15,000 kW and by the end of 30 months, the required SHP will have gone over 15,700 kW [66].

Since fouling is the most severe form of hull roughness, the ability to control this problem must rate very highly in the economic decision of which coating system to select. The choices presently available on the market vary widely but broadly fall into the following three classes:

- soluble matrix anti-fouling typified by tropical and super tropical grades
- contact leaching types generally classified as "high performance" anti-foulings
- self-polishing co-polymer anti-fouling

The unique approach to both fouling and roughness control by using the flow of water past the ship to generate a polishing action is gaining the self-polishing co-polymer anti-fouling coating increasing popularity with ship operators. It provides a smoothing benefit and simultaneously allows continuous renewal of the surface of the paint with resulting benefits in long term fouling control.

TABLE 6.1

POWERING OF 150 FOOT TRAWLER

	PARENT	10% BULB		15% BULB	
V (knots)	Effective Horsepower	Effective Horsepower	Percent Difference from Parent	Effective Horsepower	Percent Difference from Parent
9.5	156	161	+3	176	+13
10.0	184	194	+5	214	+16
10.5	216	231	+7	256	+18
11.0	263	275	+5	302	+15
11.5	340	329	-3	356	+5
12.0	457	420	-8	430	-6
12.5	595	527	-11	532	-11
13.0	717	637	-11	639	-11
13.5	826	748	-9	747	-10

[64]

FIGURE 6.1

CONFIGURATIONS OF STERN END BULBS (SEB) AND PERFORMANCE OF SAMPLE APPLICATIONS

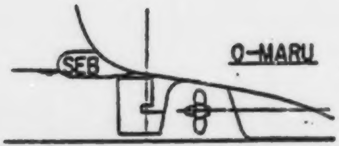
No.	1	2	3	4	5	6
Ship Sign	S-Maru	O-Maru	N-Maru	K	Oh	A*
Ship Kind	Passenger	Passenger	P-Ferry**	P-Ferry**	P-Ferry**	PCC
Route	Tokyo/ Hachijyo	Tokyo/ Ogasawara	Osaka/ Beppu	Hanshin/ Beppu	Osaka/ Shibushi	Ocean Going
G.T.	3,709 T	3,553 T	7,537 T	4,619 T	9,237 T	abt. 9,300 T
Lpp (m)	100.00 (4.000)		128.00 (3.950)	118.90 (4.000)	148.00 (4.000)	142.90 (3.60)
Bmld (m)	15.20 (0.608)		22.40 (0.691)	20.60 (0.693)	21.12 (0.571)	27.60 (0.695)
dmld design (m)	4.75 (0.190)		5.50 (0.170)	4.30 (0.145)	6.50 (0.176)	7.80 (0.197)
Cb	0.52		0.52	0.529	0.495	0.519
Rudder/Propeller	2/2	1/2	2/2	2/2	2/2	1/1
Main engine	Diesel	Diesel	Diesel	Diesel	Diesel	Diesel
MCR output (PS)	2 x 5,800	2 x 5,800	2 x 10,000	2 x 4,000	2 x 18,000	1 x 9,300
RPM	230	230	429.5	232	430	130
Original delivery	1978-4	1979-3	1973-2	1970-9	1980-2	1982-6
SEB fitting date	1981-1	1981-2	1981-2		1982-1	1982-6
*** " yard	Shimoda	Shimoda	Kawasaki, Kobe	Model test only	Koyo	Kurushima
Power saving trial	- 5.6 %	- 5.7 %	-3 to -4%	-	-	-6 to -12
Power saving in service	-	- 6.0 %	- 7.5 %	-	- 7 %	-
Power saving expected	- 5.2 %	- 4.6 %	4 %	- 4 %	- 8 %	- 4 %

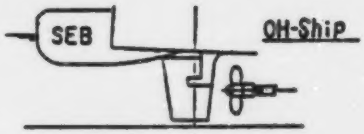
* Newbuilding ship (others are existing ships) ** P-Ferry : Passenger-Ferry

*** SEB : Supplied by KHI

( ) : Particulars of model ship

②





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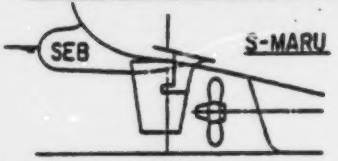
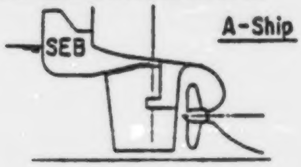
①





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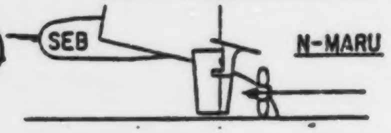

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International Paints Marine Coatings was the first company to market self-polishing coatings in 1974, in the form of their Intersmooth range. Claims of extended fouling control and surface smoothing have been substantiated by hull roughness and performance surveys carried out on many vessels at both the commencement and at the end of their service periods.

Figure 6.2 shows the results of a particular test carried out on two sister ships, A and B having a hull roughness of 139 microns and 290 microns respectively. From the two speed/power curves shown, a power requirement difference of approximately 11% can be observed [67].

The value of a self-polishing paint in smoothness control and hence its effect on hull resistance, is presented in Figure 6.3.

From the foregoing, it is obvious that self-polishing coatings, as a means of saving energy, are technically proven. Further, besides being technically sound, the use of self-polishing coatings is financially attractive. In the case of the ferries QUEEN OF VANCOUVER and QUEEN OF VICTORIA of the British Columbia Ferry Corporation it was found that the ship's hulls could be kept clean and smooth for four years while the additional first cost could be recovered within two years from savings in fuel [68].

6.4 Summary

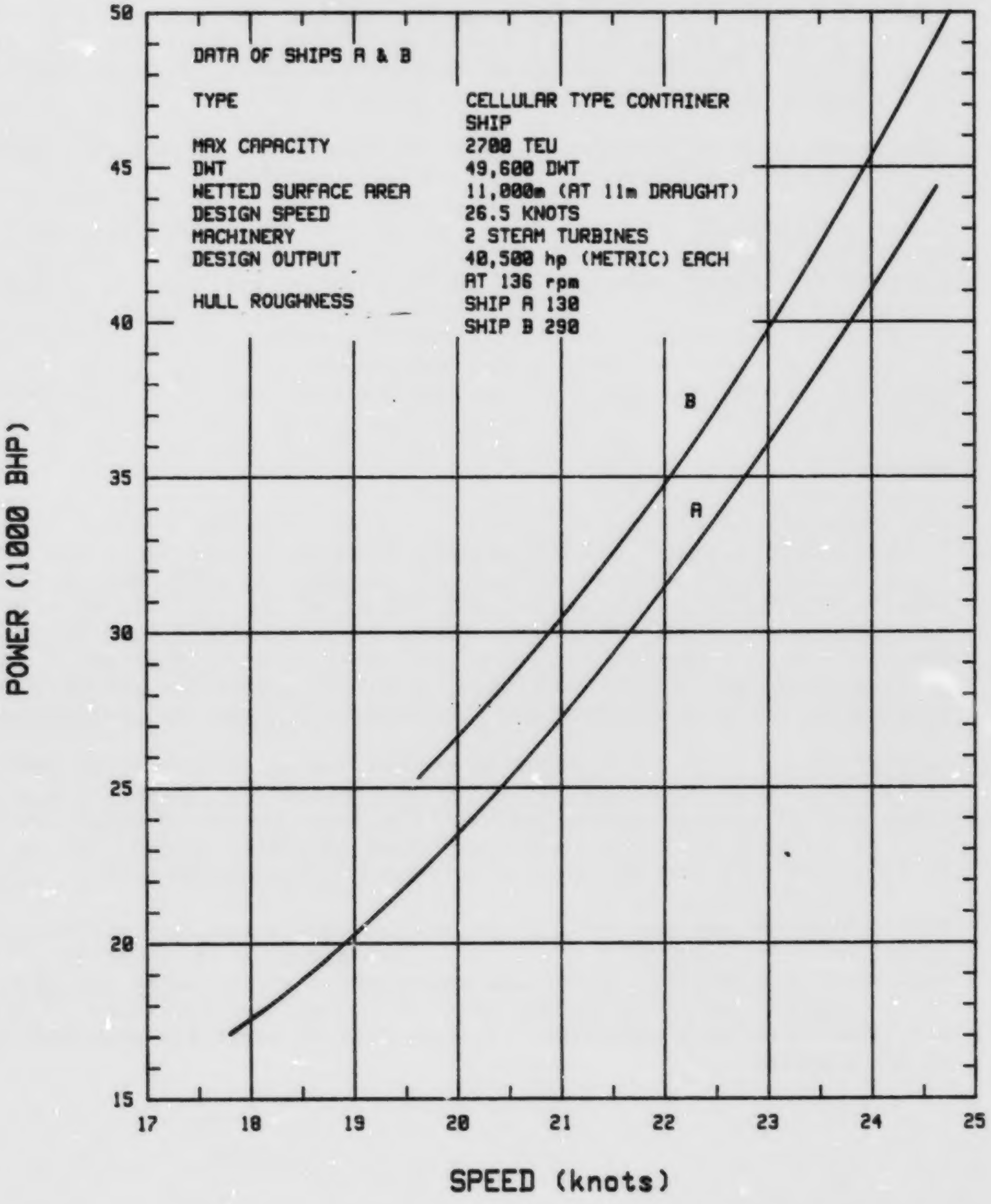
The reduction of overall resistance of a ship can be achieved by making reductions in either or both of; a) frictional resistance due to the relative motion of the hull through the surrounding water and, b) residual resistance related to the wave generating affect as the ship moves through the water and/or wake components related to water flow around the vessel's stern. Both of these aspects have been shown to greatly affect a vessel's fuel consumption. Efforts to reduce frictional resistance and to amend the wave making and wake components of resistance have been very successful.

Skin friction, which is believed to account for up to 80% of the total ship resistance, is being reduced by a variety of coatings presently available. These are a) soluble matrix anti-fouling, typified by tropical and super tropical grades, b) contact leaching types generally classified as "high performance" anti-foulings and, c) self-polishing co-polymer anti-fouling.

Modification of the wave making and wake components has been achieved by the installation of bow and stern bulbs. The objective of these bulbs is to suppress the generation of bow and stern waves, which, being formed by ship motion energy, result in wasted fuel. Bulbious bows have been in use for many years and, more recently, Professor Inui in Japan has promoted the use of stern bulbs.

FIGURE 6.2

COMPARISON OF POWER VERSUS SPEED FOR DIFFERENT HULL ROUGHNESS

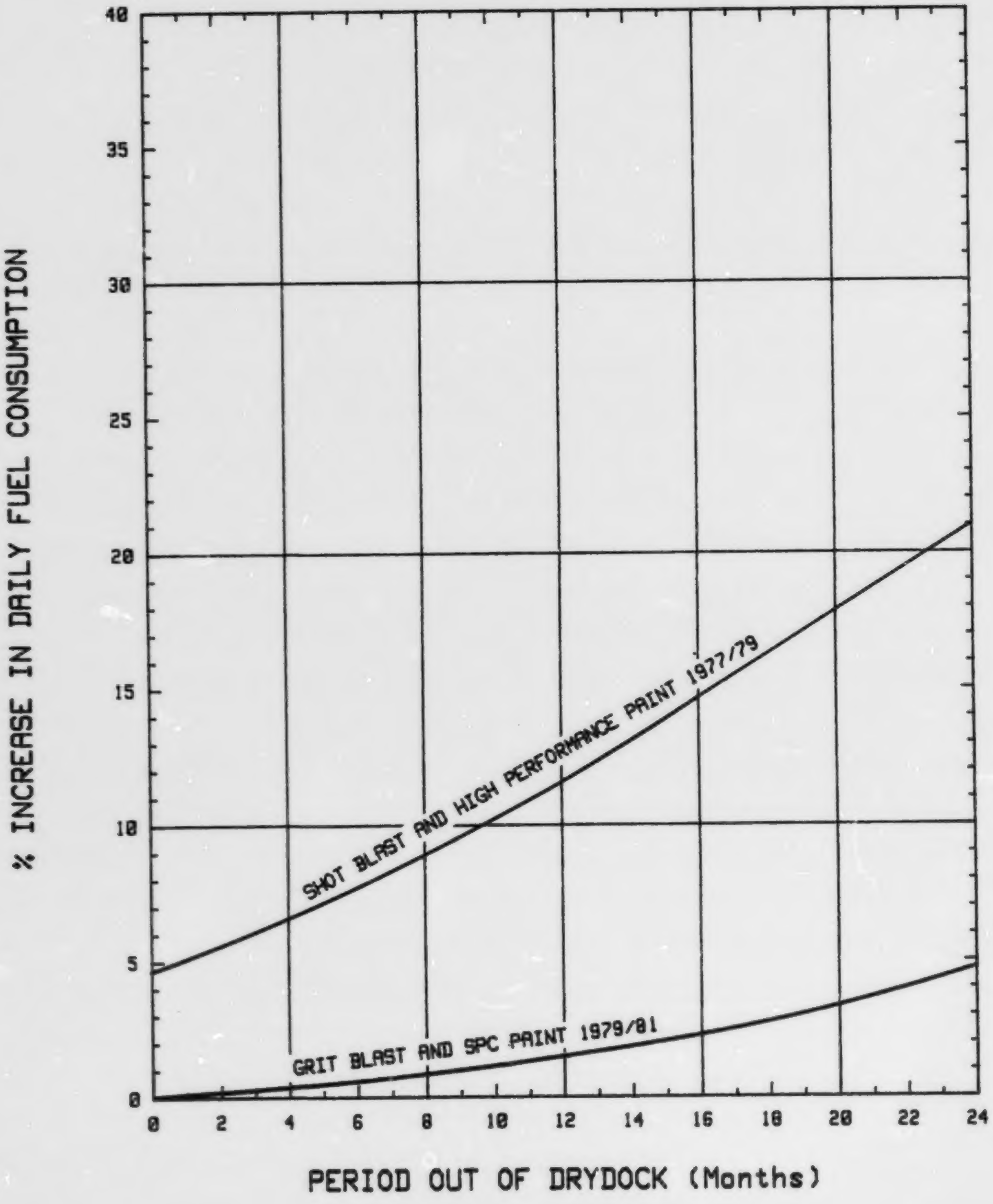


DATA OF SHIPS A & B

TYPE	CELLULAR TYPE CONTAINER SHIP
MAX CAPACITY	2700 TEU
DWT	49,600 DWT
WETTED SURFACE AREA	11,000m (AT 11m DRAUGHT)
DESIGN SPEED	26.5 KNOTS
MACHINERY	2 STEAM TURBINES
DESIGN OUTPUT	40,500 hp (METRIC) EACH AT 136 rpm
HULL ROUGHNESS	SHIP A 130 SHIP B 290

FIGURE 6.3

EFFECT OF SELF-POLISHING PAINT ON HULL RESISTANCE "AUSTRALIAN ENDEAVOUR"



* Increase in consumption is estimated for a speed of 20.5 knots

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7.0 SHIP OPERATION

7.1 General

Ship operations, as opposed to design, is of major importance in the reduction of energy consumed during a voyage. Many aspects of "operations" impact on the fuel consumption. The most important of these are discussed in this section:

- Slow speed steaming
- Ship handling
- Weather routing, and
- Draft, trim and heel conditions

7.2 Slow Speed Steaming

The escalation of fuel prices, the general decline of world trade and the resulting surplus of ships has given rise to a reassessment of ship operating costs. The ultra fast 23-25 knot container vessels built during the 1970's are now uneconomic due to their high operating costs and now operate at 19-20 knots. In the meantime, the economic speed of bulk vessels has dropped from the previous 15-16 knots to about 13 knots. Vessels which are presently under construction are designed to be suitable for the new conditions, older tonnage is slow steamed or has been re-engined.

It is well known that fuel consumption varies approximately as the cube of the speed,

$$c = kv^n$$

where:

$n = 3$ (approximately)
$k =$ constant

The index, n, depends on the shape of the ship, sea and weather conditions, etc. Figure 7.1 gives an example of such relationships for a vessel designed to make 15 knots on 60 tons of fuel per day [69].

For present day fuel costs, this correlation mainly affects the value of optimum speed, and for cargo vessels built before 1980, this speed is generally below the designed value. The optimum ship speed has been extensively researched by operators and designers taking into account the influence of fuel price, freight rates and trade patterns. It has been concluded that the optimum service speed would tend to decrease as fuel prices increase.

This section discusses the potential for:

- Replacement of propellers
- Modification of diesel engines
- Modification of steam turbines, and
- Modification of boilers

7.2.1 Replacement of Propellers

A fixed pitch propeller can be designed to satisfy only one set of performance conditions and it is therefore accepted practice to optimize the propeller to suit loaded conditions at the service or maximum power of the machinery. It follows that if the propeller and ship are operating at conditions removed from the original design point for extended periods, it may be economically advantageous to replace the propeller with one designed to operate in accordance with the new operating speed of ship and engine [70, 75].

As most vessels or classes of vessels are ordered with a spare propeller, this propeller could be supplied for a different operating speed and if the optimum ship speed continues to fall, as predicted, it could be economic to replace the original propeller by the spare.

The installation of a variable pitch propeller, although initially more expensive and generally less efficient than a fixed pitch propeller, can be advantageous for vessels which are required to operate at a variety of different speeds. The use of a controllable pitch propeller would not be economic on a long term basis for long voyage deep sea vessels. A full discussion of propeller efficiency is provided in Section 4.0.

7.2.2 Diesel Engines

The specific fuel consumption versus engine load characteristics for the modern diesel engine is essentially flat for both medium speed and slow speed engines. However, as the exhaust gases from the medium speed engine have a higher temperature than those from the slow speed engine, the quantity of waste heat which can be recovered from the medium speed engine at part load is greater than for the slow speed machine. If this lost production of steam or heat from the waste heat recovery system when slow steaming must be compensated for by direct firing of a boiler or the operation of an auxiliary engine for electrical power generation, the all purpose fuel consumption for the vessel will increase considerably and some of the advantages of slow speed steaming will be lost.

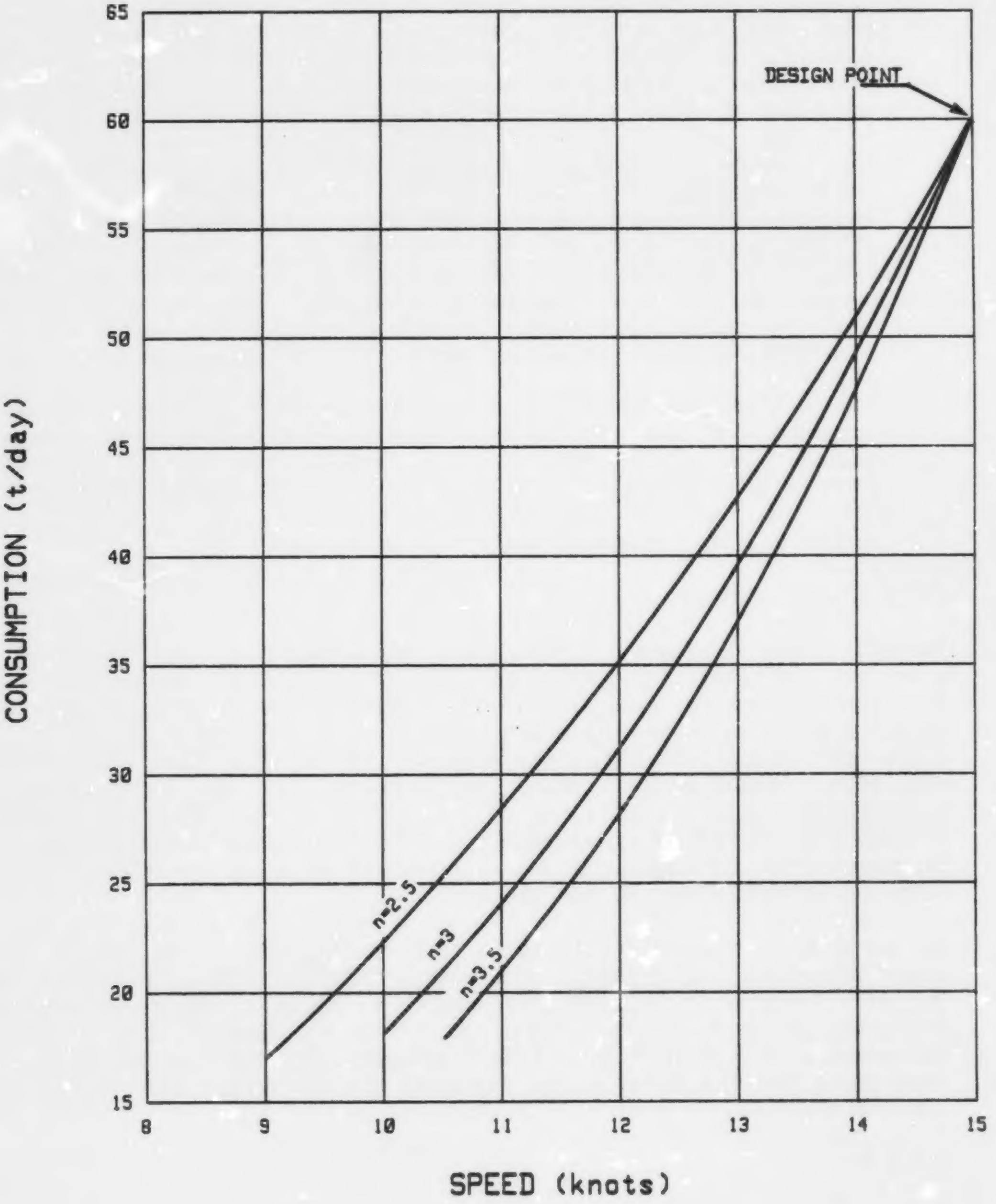
Similarly if a fixed pitch propeller system is operated at off design conditions, engine power take offs to constant speed machines such as alternators cannot be used and auxiliary engines must be operated in their place. Propulsion systems fitted with controllable pitch propellers which can operate at variable load with the engine at constant speed have considerable advantage in this case.

To improve the specific fuel consumption at low loads, diesel engines should be modified as follows:

- The fuel system should be modified to allow injection timing to be varied. The Sulzer Variable Injection Timing (VIT), the Bosch fuel pump with a modified edge or the electronic fuel injection system are specifically designed for this purpose [71].

FIGURE 7.1

SHIP'S FUEL CONSUMPTION RELATED TO SPEED



[69]

- Provision must be made to keep the scavenging air temperature high and thus maintain the exhaust temperature at normal operation levels. In extreme cases this can be done by preheating the air before the turbo-charger but in most instances adjustment of the cooling water flow through the charge air cooler is adequate.
- "Slow" injection nozzles, when running continuously, i.e., for more than 24 hours, at reduced load (below 40% MCR) can be advantageous both with respect to economics and engine cleanliness. These nozzles have a modified fuel spray geometry which improves burning at low load [72].
- For low power steaming of engines fitted with constant pressure turbo charging, auxiliary electric motor driven scavenge air blowers must be used below 40% MCR load. If prolonged steaming at off design conditions are required the turbo charger inlet nozzle ring should be changed to suit the new gas flow conditions. Some modern engines are now being supplied with variable nozzle geometry turbo chargers. The reintroduction of under piston supercharging would be advantageous for slow speed operation.
- The use of a camshaft fitted with two sets of cams has been proposed for use with highly supercharged medium speed diesel engine. One set of cams would be designed for normal operation, with large overlaps of inlet and exhaust valve opening times while the other set for low load operation with small overlaps. Up until now, this proposal has not been adopted [73].

7.2.3 Steam Turbines

The efficiency of steam turbines when operating at off design conditions is poor and considerable advantages can be gained by modifying the high pressure turbine if slow steaming is required for an extended period. In recent times many owners have preferred to modify the high pressure turbine, gears and propellers in preference to re-engining and have shown the resulting reduction in fuel consumption to be substantial. The modifications will usually be limited to the high pressure turbine and include replacement of inlet nozzles, throttle valves and diaphragm nozzles throughout the turbine in order to reduce flow area. At times it has been found advantageous to reblade the turbine [74].

The purpose of the modifications is to eliminate throttling losses, increase stage pressures and optimize the energy distribution within the turbine. The efficiency gain is dependent upon the degree of derating and must be evaluated for each specific case. At 50% power, overall specific fuel rate improvement of 1.0 to 4.5% can be achieved. The turbine modifications are reversible and the fully rated capacity of the system can be restored at some future time, if desired [75, 76].

7.2.4 Boilers

Boiler efficiency generally remains high throughout the load range and often increases with decreasing load as is shown in Figure 3.10, but the superheated steam outlet temperature generally decreases below the controlled temperature point for high temperature plants. The lower superheat steam temperature decreases the energy available to the turbine and leads to a loss of efficiency. Whatever can be done to maintain the steam temperature as close to the design values as possible will improve the part load efficiency.

In a two boiler ship operating at low powers, it is always more efficient to operate with one boiler wherever possible. This type of operation will provide, in general, higher steam temperatures which will improve plant efficiency. The overall plant efficiency will also increase since many of the auxiliaries of the non-operating boiler may be shut down. Keeping the inactive boiler hot will minimise corrosion and also allow it to be quickly returned to service in a need of high power or manoeuvring.

Great care must be taken to maintain the flue gas temperatures above sulphuric acid dew point which will be lower at part load. The flue gas temperature must be therefore controlled by use, for example, of a regenerative air heater or feed water temperature in the economiser [32].

7.3 Ship Handling

The navigation of a ship has traditionally been by straight runs coupled with changes of track, at appropriate points, necessary to bring the ship to its destination. For the majority of deep sea voyages, the main concern is to ensure that the ship follows its intended track with minimum deviation. To satisfy this need, the autopilot was introduced many years ago as a means to control steering and release a man for other duties.

It has long been recognised that an experienced helmsman is far more energy efficient than a simple autopilot. The control system for the autopilot is reactive and non anticipating, this tends to cause frequent small changes of helm to occur which increases drag and hence increases fuel consumption.

The increase of drag due to changes in helm is high and affected by:

- rudder drag as a longitudinal component of a force generated on the turned rudder
- inertial drag as a longitudinal component of the inertial force produced by the turning of the ship with accompanied mass of water

These forces, shown in Figure 7.2, indicate that the way the ship is steered has a large effect on total power demand. The penalty in bad cases has been quantified as a 5 - 10% reduction in propulsive performance. This can be reduced by the use of a more sophisticated autopilot [79].

Most autopilot manufacturers use a control system based on angular displacement to control the path of the ship, however, a more appropriate means of control is to monitor the angular acceleration and to use it as feedback to control the steering gear. This system permits more precise control of the ship and the change of course can be detected earlier and hence drag is reduced as smaller angles of helm are required for correction.

A more sophisticated steering device is the self-adaptive autopilot. This system finds the optimum setting which will control a ship so that its course and rudder deviations are at a minimum. An accurate mathematical model of the ship's course keeping capability is required and input parameters such as speed, rudder angle, rate of turn etc. are fed into the computer which will then calculate the rudder angle or force required to maintain the vessel on course [80].

The extra distance travelled by a ship is the product of an irregular course, but fuel losses due to this are small compared to the propulsion losses due to rudder drag. Some of the latest models of autopilots are extremely effective during rough seas, as the computer continuously monitors the actual movement of the ship, compares it with the prediction of the model and can discriminate deviations caused by weather and apply a necessary rudder movement. Characteristics of the Mitsubishi TONAC PILOT I having such qualities are shown in Figure 7.3 [81].

7.4 Weather Routing

Weather routing has been advocated for some time as a method of reducing fuel costs. A form of weather routing has been in common practice for many years for North Atlantic passages, i.e. the shorter northern passage in summer and the longer southern route in the winter months. The present approach is much more sophisticated and has recently culminated in the development of an algorithm which combines weather and wave forecasting with vessel seakeeping characteristics in conjunction with operating costs. The resulting model, when solved by a dynamic programming technique, will identify the optimum vessel routing consistent with prevailing weather and operational constraints, as well as related economic factors.

Therefore, new ships can be provided with information by a "Weather Routing Company" to save both time and fuel. Actual voyage simulations can be performed on the "Routing Company" computer and studies of alternative routes across the weather data base made to determine optimum voyage times and distances.

In order to compare the optimal routings with the actual vessel track, an analysis has been made of the noon position records of the logbook and a time history of vessel position was established [82]. A comparison of costs predicted by the program for the optimum route, and the actual route has formed the basis for establishing the potential benefit of the algorithm. The results for North Atlantic voyages shown in Figure 7.4 dramatically reveal the potential benefits to be gained from weather routing.

For a container ship the "Optimised Ship Routing" (OSR) algorithm appears to offer an advantage on the order of $3,000 to $3,500 per voyage. The westbound voyages were particularly good opportunities for energy conservation with savings of 4 to 7% in fuel costs.

Weather routing is in common use throughout the world as weather maps are readily available from satellite imagery. Vessels use this system to protect cargo from damage or to achieve the most energy efficient voyage.

FIGURE 7.2

FORCES ACTING ON SHIP DURING TURN

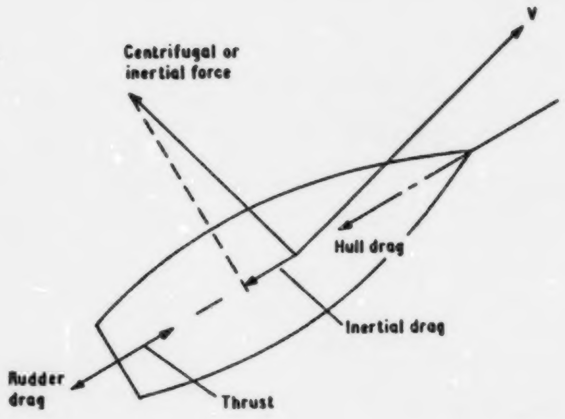

V
Centrifugal or
inertial force
Hull drag
Inertial drag
Rudder
drag
Thrust

Table I: Ratio of rudder drag to total drag

RUDDER ANGLE (deg)	CARGO SHIP R_R/R_0 (%)	TANKER R_R/R_0 (%)
5	0.92	0.76
10	3.68	3.02
15	8.28	6.80
20	14.73	12.08

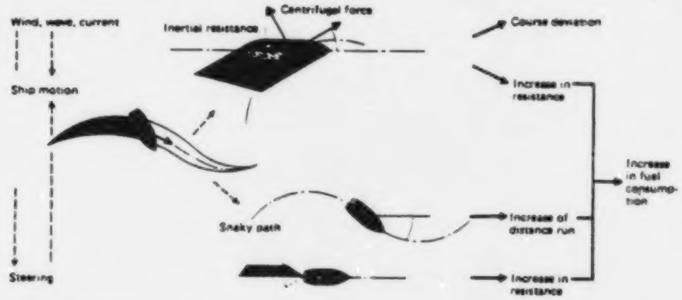
Table II: Ratio of inertial drag to total drag

RUDDER ANGLE (deg)	CARGO SHIP R_I/R_0 (%)	TANKER R_I/R_0 (%)
5	22.2	9
10	48.7	29.7
15	75.4	55.4
20	105	88.3

[79]

FIGURE 7.3

FACTORS RESPONSIBLE FOR INCREASE IN FUEL CONSUMPTION IN RELATION TO HORIZONTAL MOTION OF SHIP


Wind, wave, current
Inertial resistance
Centrifugal force
Course deviation
Ship motion
Increase in resistance
Increase in fuel consumption
Snaky path
Increase of distance run
Steering
Increase in resistance

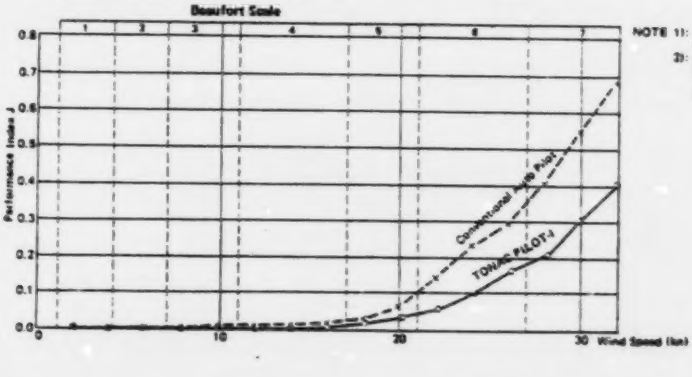
In order to provide an all-round solution to these problems, the TONAC PILOT-I incorporates the following functions:

Reduce unnecessary yaw
Remove high-frequency disturbance (wave) causing unnecessary yaw → Disturbance filter
Self-adjustment according to sea conditions

Minimize inertial resistance due to yaw and sway
Minimize increase of distance run due to yaw and sway
Minimize increase in resistance due to steering
→ Performance index indicating overall increase in fuel consumption
- Steering control to optimize the performance index
- Self-adjustment of control parameters according to ship and sea conditions

→ Monitor and judge necessity of the above adjustment

PERFORMANCE COMPARISON OF CONVENTIONAL AUTO PILOT
TO MITSUBISHI TONAC PILOT-I

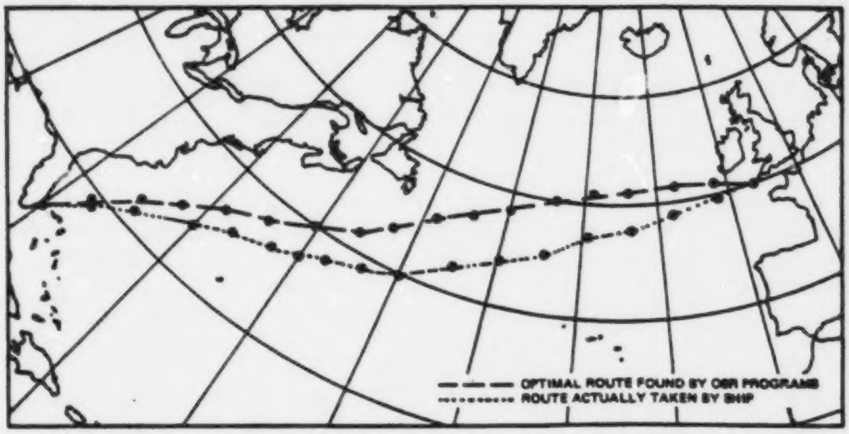


NOTE 1): VLCC, Full Load,
Vs = 16kn
2): $J = (E - E_o)/E_o$
E = Actual propulsive energy
E_o = Ideal propulsive energy

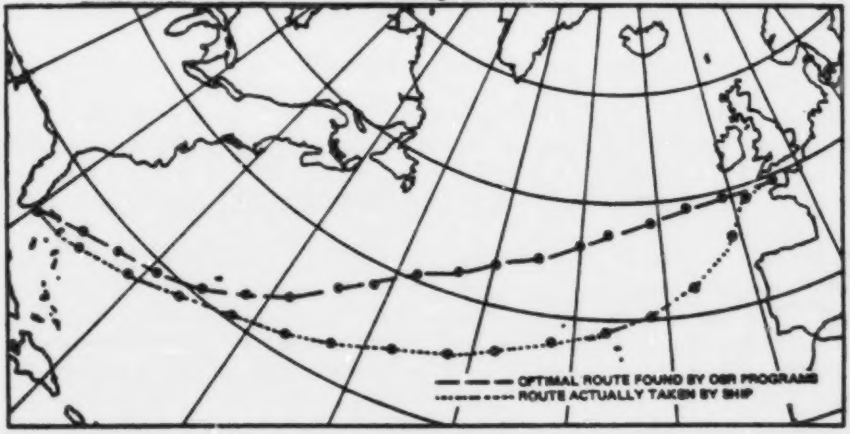
[81]

FIGURE 7.4

BENEFITS GAINED FROM WEATHER ROUTING


OPTIMAL ROUTE FOUND BY OSR PROGRAMS
ROUTE ACTUALLY TAKEN BY SHIP

Alternate routes for March westbound voyage: Bremerhaven to Port Haven to Port Everglades, 3/14/80 through 3/23/80


OPTIMAL ROUTE FOUND BY OSR PROGRAMS
ROUTE ACTUALLY TAKEN BY SHIP

Alternate routes for eastbound voyage: Port Everglades to Rotterdam, 4/1/80 through 4/9/80

Distances (nm) for alternate routes

	OSR Route	Logged Route
March/eastbound	3487	3552
April/eastbound	3576	3601
March/westbound	3682	3903
April/westbound	3639	3909

Steaming times (hr) on alternate routes

	Scheduled	OSR Route	Optimized Logbook Route	Actual Voyage
March/eastbound	180.3	164.7	164.7	170.5
April/eastbound	185.0	177.9	187.1	176.8
March/westbound	184.8	186.1	196	196.2
April/westbound	184.8	188.6	196	205.6

Fuel consumption (barrels) on alternate routes

	OSR Route	Optimization of Logged Route
March/eastbound	6375	6508
April/eastbound	6169	6177
March/westbound	7303	7623
April/westbound	7314	7849

[82]

7.5 Draft, Trim and Heel Control

Fuel consumption has proved to be an essential feedback in finding optimum trim under different sea and load conditions. Optimum trim has a direct effect on the hull resistance and speed through the water. All changes of this condition have effects on fuel consumption and only close monitoring can assure the best trim setting at all times.

Many such control and monitoring systems exist today, and are being improved with respect to their accuracy and effectiveness. The functions of one of those systems is shown in Figure 7.5 [83].

Stabilizers and hence control and reduction of ship's resistance by the elimination of rolling can significantly reduce the fuel consumption. To estimate the fuel savings, the sea conditions likely to be encountered by the vessel must be evaluated. In the North Atlantic, for example, state 6 seas occur 15% of the time and the gains provided by stabilizers fitted to fine lined vessels can be substantial.

Two types of stabilizers are in common use, the "tank type" are much cheaper and simpler than the "fin type" but they require more space and reduce the earning capacity of the vessel. However, development in the stabilizer field is very rapid and the types produced today may be obsolete in a few years.

7.6 Summary

There are many options open to vessel owners to improve fuel consumption by effective decision-making systems supported by detailed measurement and monitoring of ship performance and its environment.

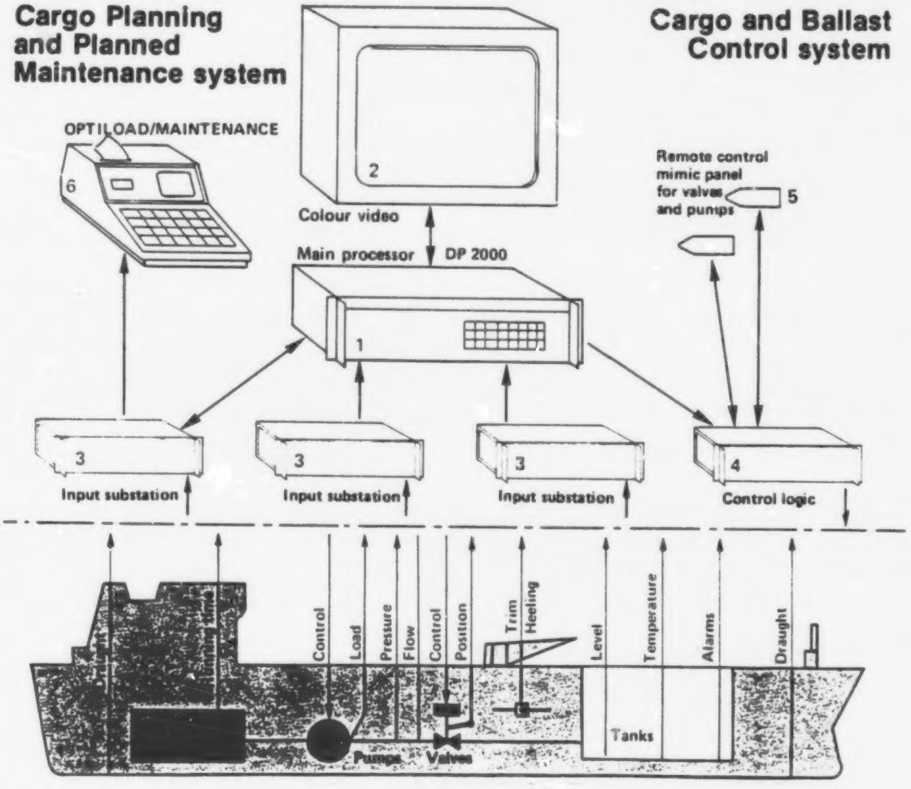
The easiest means in this category of reducing overall fuel consumption is slow steaming, especially for operators not sailing to a liner schedule. The magnitude of slow steaming benefits depend also upon propulsion machinery, that is, whether the design is adapted to perform efficiently at lower loads. The efficiency of prime movers and propellers when operating at off-design conditions is usually poor. By introducing modifications, maximum efficiency can be matched with optimum speed. For example, it could be; replacement of the propeller with one better suited for the new speed; application of variable injection timing; or special injection nozzles in diesel engines, etc.

Another area for influencing fuel consumption is improved ship handling; to reduce additional resistance resulting from excessive rudder action; and accurate navigaton, to minimize extra distance travelled. To achieve this, the application of modern navigation aids is essential. Maintaining optimum draft and trim through careful monitoring and accurate calculations can also have a positive effect on fuel consumption.

Finally, weather routing has become a remarkable element in fuel economy, mainly due to the development of computers and satelites. Specialists in weather routing utilizing all available meteorological data, computers, and specific information about a vessel, can provide timely recommendations to a master and enhance safety and fuel performance.

FIGURE 7.5

CARGO AND BALLAST CONTROL SYSTEM


Cargo Planning
and Planned
Maintenance system
Cargo and Ballast
Control system
OPTILOAD/MAINTENANCE
6
2
Colour video
Remote control
mimic panel
for valves
and pumps
5
Main processor
DP 2000
1
3
Input substation
3
Input substation
3
Input substation
4
Control logic
Running time
Control
Load
Pressure
Flow
Control
Position
Trim
Heeling
Level
Temperature
Alarms
Draught
Pumps
Valves
Tanks

[83]

8.0 APPLICATION OF FUEL EFFICIENCY MEASURES TO VARIOUS CANADIAN SHIPS

8.1 General

The application of fuel efficiency measures to Canadian ships must reflect the special operating requirements of those ships and the modern trends now prevailing around the world. This section discusses those efficiency measures by:

- identifying trends in the worldwide fleet
- reviewing the Canadian fleet, and
- assessing the potential improvements in Canadian Great Lakes bulk carriers, ferries, ocean going ships and coastal ships.

8.2 Trends in the Worldwide Fleet

Due to the events of recent years shipowners and operators have become even more aware of the influence of fuel prices on transportation costs. Hence there is considerable interest in building fuel efficient ships and developing more efficient engines and machinery systems. As part of the effort to remain competitive and contain any increases in ship operating costs, owners are becoming willing to allow their ships to become more capital intensive. Some of the ways in which this is exhibited include:

- designing ships for a specific trade, i.e., container ships, Ro-Ro ships, etc.
- designing larger ships
- designing ships where the initial capacity is larger than immediately required in anticipation of future needs
- incorporating features and systems which are favourable in reducing both ship operation and maintenance costs.

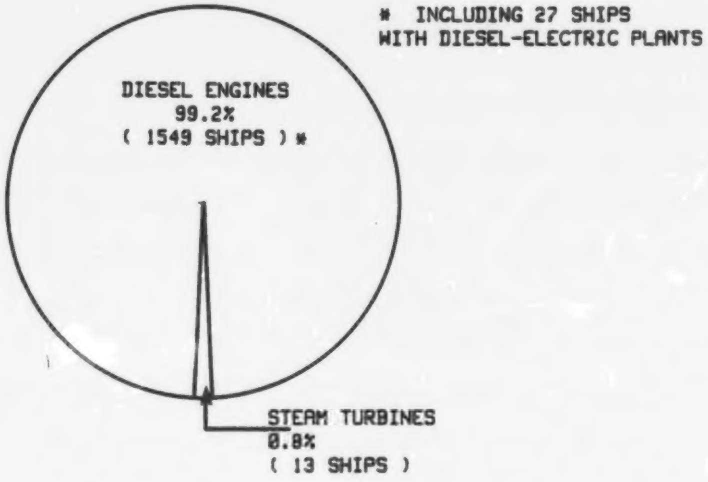
Historically, obtaining the optimum specific fuel consumption from a selected propulsion plant design has not taken the highest priority. It was only with the unprecedented rise in fuel oil prices in the 1970's and with the growing awareness of the rapid depletion in the readily available supplies of oil that the need for fuel economy became great enough to justify the inconveniences and expense of more sophisticated ships. That is why the previously popular but less fuel efficient engines, such as steam turbines, are no longer installed today, except in special circumstances. The U.S.A. has remained the last bastion of steam turbine propulsion until recently, but is now changing to more fuel efficient prime movers for new tonnage. Recently, three large container ships have entered into service powered by the first low speed diesel engines to be constructed in the U.S.A. for 30 years.

The simplest method of analysing the recent trends in shipowners' demands is by inspection of recent contracts held by shipyards throughout the world. Figures 8.1 to 8.4 and Table 8.1 have been prepared according to data published by "Motor Ship" and shows present worldwide shipyard contracts [97]. It can be seen that only 13 ships with steam turbines have been

FIGURE 8.1

SHIPS ON ORDER IN OCTOBER 1983
(MERCHANT SHIPS OF 2000 DWT AND ABOVE)

MAIN PROPULSION PLANT



TYPE OF DIESEL ENGINES

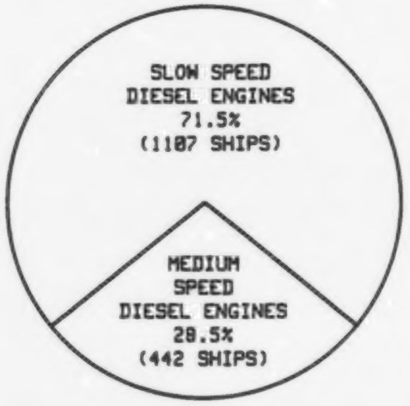


FIGURE 8.2

SHIPS ON ORDER IN OCTOBER 1983
(MERCHANT SHIPS OF 2000 DWT AND ABOVE)

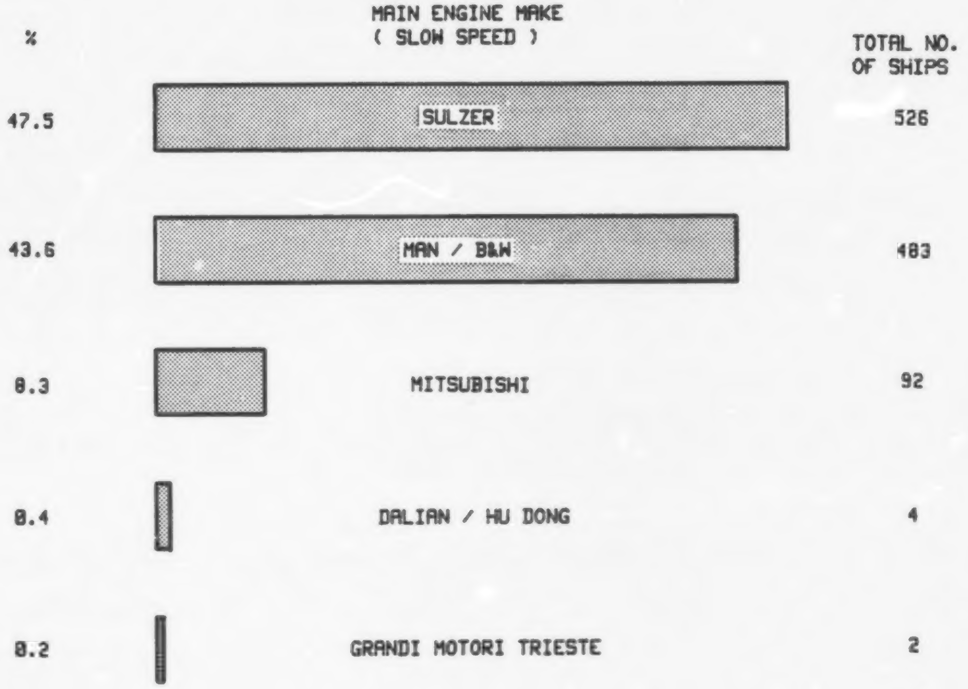


FIGURE 8.3

SHIPS ON ORDER IN OCTOBER 1983
(MERCHANT SHIPS OF 2000 DWT AND ABOVE)

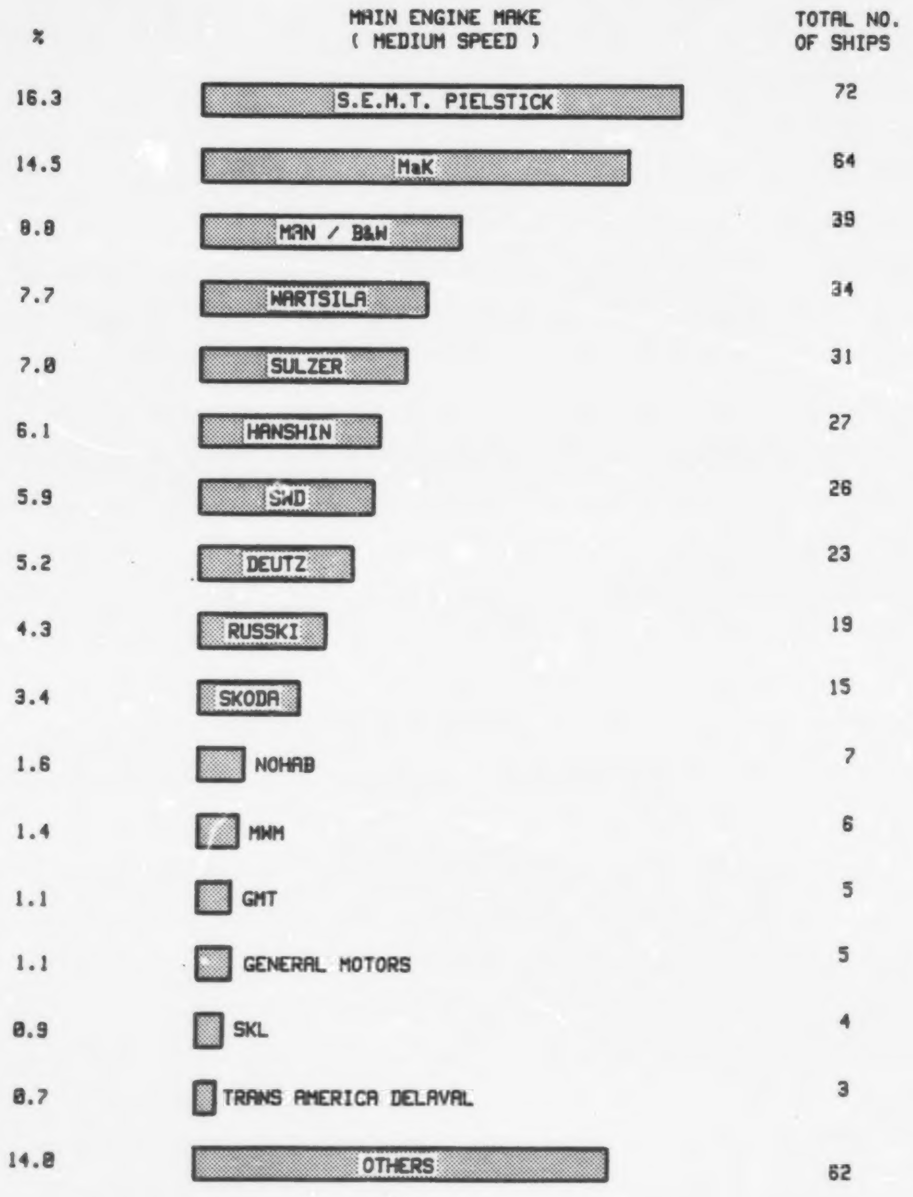
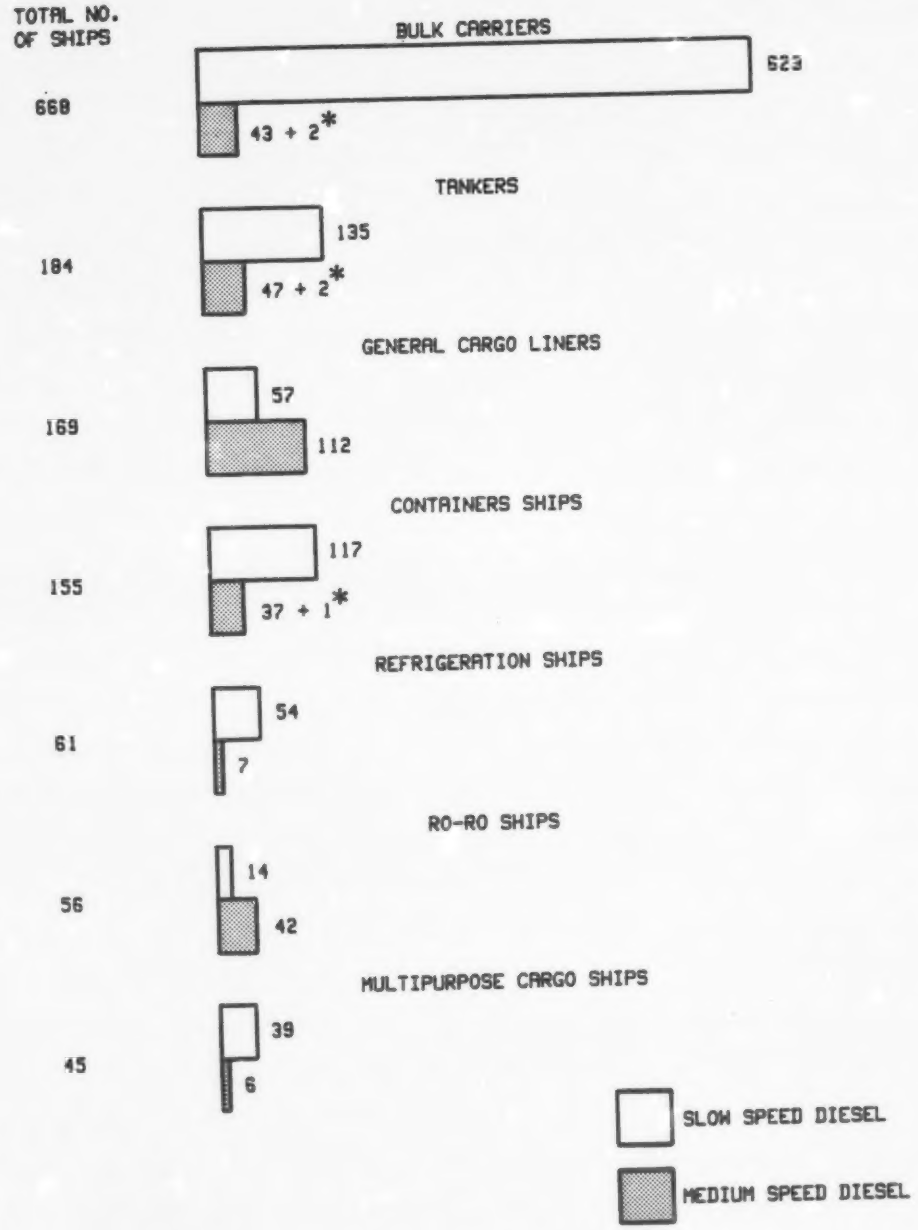


FIGURE 8.4

SHIPS ON ORDER IN OCTOBER 1983
(MERCHANT SHIPS OF 2000 DWT AND ABOVE)



* STEAM TURBINES

MELVILLE SHIPPING LTD

TABLE 8.1

SHIPS ON ORDER IN OCTOBER 1983
(MERCHANT SHIPS OF 2000 DWT AND ABOVE)
(SHIP TYPES OTHER THAN IN FIG. 8.4)

TOTAL NO. OF SHIPS	TYPE OF SHIP		WITH SLOW SPEED DIESEL	WITH MEDIUM SPEED DIESEL
28	Passenger/Car Ferry		2	26
23	Car Carrier		21	2
18	LPG Tanker		12	6
15	Dredger	(1)	-	13
14	Icebreaker	(2)	-	6
13	Coaster		3	10
13	Heavy Lift		8	5
11	Barge Carrier	(3)	-	10
11	Drilling Rig	(4)	-	6
10	Rail Ferry		8	2
10	Research Vessel		-	10
8	Offshore Supply		-	8
6	LNG Tanker	(5)	-	1
6	Trawler		3	3
5	Tug		-	5
5	Cruise Liner	(6)	1	3
5	Crane Vessel	(7)		1
2	Cable	(8)		1
20	Other Special Vessels	(9)	-	10

Remarks
(1) + 2 Diesel Electric Main Propulsion
(2) + 7 Diesel Electric Main Propulsion + 1 Nuclear/Turbine
(3) + 1 Nuclear/Turbine
(4) + 5 Diesel Electric Main Propulsion
(5) + 5 Steam Turbines
(6) + 1 Steam Turbines
(7) + 4 Diesel Electric Main Propulsion
(8) + 1 Diesel Electric Main Propulsion
(9) + 10 Diesel Electric Main Propulsion

ordered recently, of which five are LNG ships. The figures also give a clear view of the shipowners' preference for slow speed diesel engines over medium speed engines, and the popularity of Sulzer Bros. as the world's main supplier of ship propulsion engines. Until recently more marine engines were provided by the MAN/B&W group of companies. Recent developments by Sulzer Bros. of the "super long stroke" engine, which provides the most fuel efficient diesel propulsion system presently available, has enabled that company to capture the largest share of a shrinking market.

Shipowners' intentions to conserve energy are reflected in the continuous reduction of ship propulsion power and confirmed by an analysis of shipyards' contracts. In October 1983 the average horsepower of the ships on order was 7,494 kW, i.e., 733 kW less than 3 months previous.

Shipowners are also improving the efficiency of their fleets by retrofitting more efficient engines to existing ships. The most expensive conversion is the replacement of turbine machinery by diesel engines which, as a general rule, can give approximately 30% saving of fuel cost. The decision to re-engine from steam turbines to diesel and even from diesel to diesel has been taken by many shipowners. In addition to re-engining, some vessels have had associated characteristics changed at the same time, i.e., twin to single screw, a reduction in overall power, an increase of cargo capacity by "jumboising", etc.

The steam to diesel engine conversion is the most beneficial when the power exceeds 22,000 kW and for this reason there has been few re-enginings of cargo ships with lower power - a sign that owners of these ships have been unable to justify the cost. However, many re-engining feasibility studies have been carried out which have shown that the payback would occur within a reasonable period of time, much less than the remaining life of the ship, but the financial returns are very sensitive to fuel oil prices. Instead of re-engining, many operators of high powered steamships have chosen only to readjust the thermal balance of the steam plant; this has given them appreciable fuel savings. Finally, substantial savings in the fuel bill have been made by retrofitting engines or modifying existing engines to burn cheaper heavy fuel oil instead of diesel oil.

8.3 The Canadian Merchant Fleet

Although there is a world wide impetus to conserve energy, this has had a limited impact upon Canadian marine transportation. Canada requires a large variety of vessels to service its needs but the number of Canadian vessels supplying those services are few and many are old and energy wasteful. The Canadian shipowners' views, with some notable exceptions, in the fields of ship design and energy efficiency, are seen to be several years behind the trends of most foreign progressive owners. In addition, the Government does not provide any incentives to the shipowner to invest in fuel efficient ships or to improve the efficiency of existing older tonnage.

The Canadian contemporary merchant fleet is not fuel efficient. Figure 8.5 shows a breakdown of the types of propulsion engines fitted to Canadian ships [98, 99]. Over 25% of ships fitted with steam popuision engines are inland waterway vessels that have extended their life far in excess of the world's average. Some vessels built before World War I, which are powered by steam reciprocating engines or steam turbines, are still in operation. The longevity of these vessels is attributable to the lack of wastage of their hulls when operating in the fresh water of the Great Lakes.

While it cannot be economically justified to invest substantial capital in older tonnage which has limited further use, considerable savings of fuel can be achieved by improvements to steamers which have more than ten years further service life. Twenty-three steam ships, which mainly operate on the Great Lakes, are relatively modern vessels and may profit from investment to reduce their consumption of fuel. Unfortunately the operating season in the Lakes is relatively short, therefore the payback period on any investment in this respect tends to be longer than normally considered acceptable.

Although oil fuel prices have stabilized for the meantime, prices will probably increase faster than inflation in the future and marine turbine plants will be at a further disadvantage. Therefore, any new buildings should employ as many fuel efficiency measures as are proven effective if the ship is to be competitive throughout its life. Particular features which help to conserve energy should be selected to match the operating profile of the ships. Any study of the available energy efficiency measures and technology must take into account the particular operating requirement of each ship type.

8.4 Great Lakes Bulk Carriers

Ships which navigate in the Great Lakes are principally bulk carriers, developed over many years to transport raw materials in the most economical manner possible. In general the principal dimensions of the hull are determined by the size of the Seaway locks and other channels. Due to the preoccupation of shipowners to carry the maximum volume of cargo, the laker is therefore of extended length compared with a conventional deep sea merchant vessel of the same deadweight and the hull form hydrodynamically inefficient.

The average age of the Canadian Great Lakes fleet is 28 years and some ships are almost 80 years old [98, 99, 101]. There is nothing inherently wrong in this, provided the ships do not waste fuel. Unfortunately, in most instances, age equates to inefficiency. It is generally claimed and accepted that steam turbine propulsion for this size of vessel is completely uneconomical today and virtually obsolete. Moreover, due to the variable power profile of lakers steam propulsion is even more inefficient.

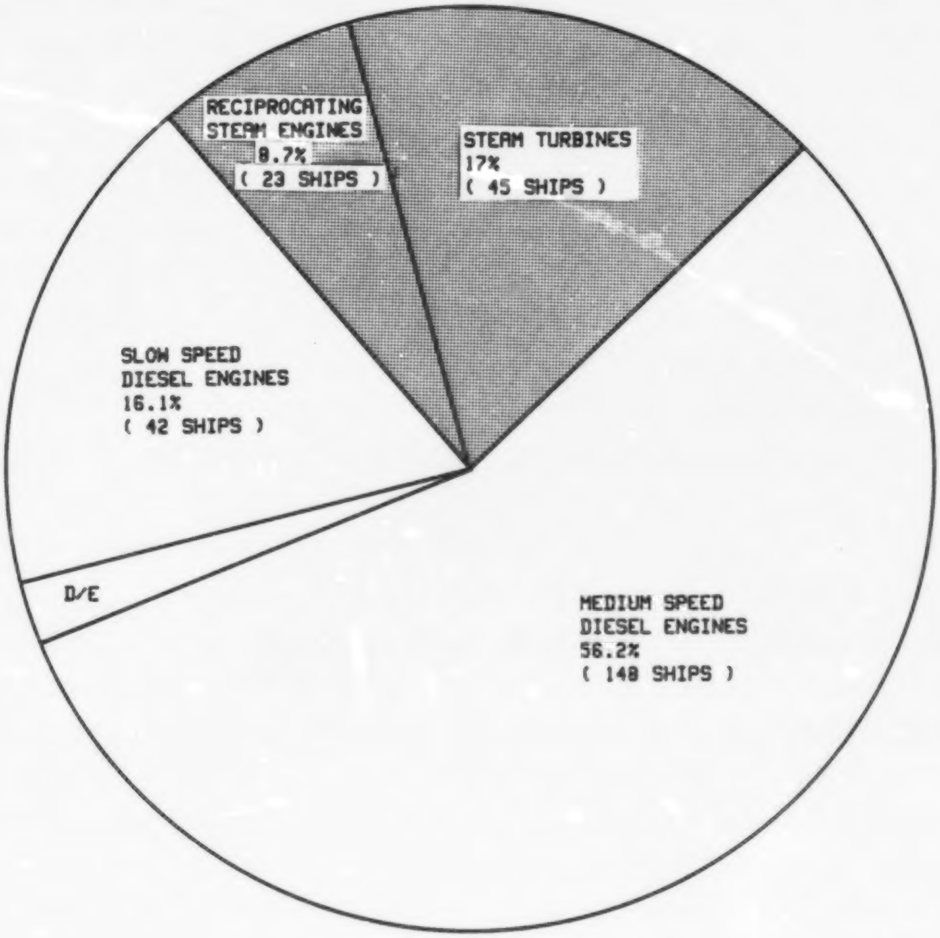
From Figure 8.6 it may be seen that half of the Canadian Great Lakes fleet is powered by steam. Long periods of low load operation make these ships extremely inefficient. To make matters worse, some steamers burn diesel oil in their main boilers.

8.4.1 Operating Profile

Propulsion machinery for this class of vessel must meet stringent requirements. Operation in the Seaway and the Lakes requires a substantial

FIGURE 8.5

THE CANADIAN MERCHANT FLEET IN JANUARY 1983
(1000 GRT AND ABOVE)

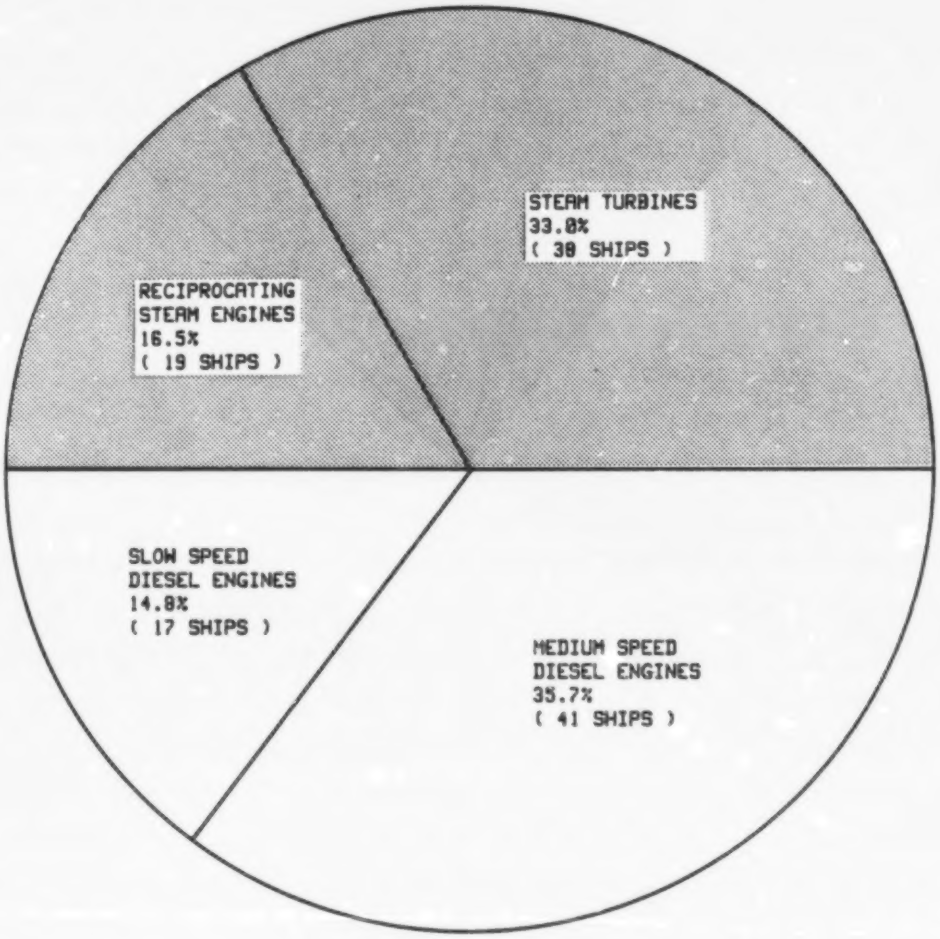


D/E - DIESEL/ELECTRIC MOTOR PROPULSION 1.9% (5 SHIPS)

[98] [99]

FIGURE 8.6

THE CANADIAN GREAT LAKES BULK CARRIER FLEET
IN JANUARY 1983 (1000 GRT AND ABOVE)



amount of manoeuvring, long periods at idle and sudden load changes of the engines which requires the machinery to be flexible and may result in high rates of wear of the propulsion plant. A typical voyage profile is shown by Table 8.2 [100]. This shows a part section of a manoeuvring table for a large laker and illustrates the operation of the engines at off-design conditions. The table indicates a considerable period of time spent at low load, which has a marked affect on the specific fuel consumption of the vessel.

Table 8.3 is based upon data obtained from shipowners and from technical journals. The table shows the essential characteristics of ship operation in the Seaway and the Great Lakes. Table 8.4, the operating profile of a typical Great Lakes bulk carrier, was derived from Table 8.3.

8.4.2 Potential Saving from the Existing Great Lakes Fleet

An attempt has been made to quantify the annual fuel cost penalty paid by lakers. The assumptions used in the estimate are given in Table 8.5 and details of the calculation in Table 8.6. The calculations indicate that approximately U.S. $51.1 million is lost annually by Canadian companies in the operation of fuel inefficient ships on the Great Lakes.

8.4.3 Energy Efficiency Measures for New Vessels

Forecasts predict very slow growth in marine bulk cargo traffic on the Great Lakes [102] in the next few years and very few newbuilding projects are expected within the next five years. However, from consideration of the typical operating profile of Great Lakes ships, the following system is considered the most fuel efficient for new vessels, but may slightly increase the capital cost.

- Main Engine
  - . 2 medium speed diesel engines sharing the required propulsion power in ratio 75/25% (the so called "father and son" system).

- Electrical Energy Generation:
  - . the smaller main engine driving one generator to supply the self-unloading gear when in port
  - . a turbo-generator, supplied with steam from an exhaust gas waste heat recovery boiler, or a generator driven from a propulsion engine power take-off for electric supplies when underway
  - . one auxiliary diesel engine driven generator for harbour service use and standby services

To illustrate how this system would operate, the vessel ATLANTIC SUPERIOR has been selected as an example:

- Propulsion Power	8200 kW
- Electrical Load at Sea	350 kW
- Electrical Load in Port During Ballasting Operations	600 kW
- Electrical Load in Port During Unloading Operations	1900 kW
- Bow Thruster Power	900 kW

TABLE 8.2

MAIN ENGINE MOVEMENTS THROUGH SEAWAY AND WELLAND CANAL

SECTION I	Montreal to Iroquois Locks (the last lock in the Seaway System)	
	Total Time Taken	17 hours 24 minutes
	Total Movements	204
	Power Variation % During Manoeuvring	Ahead 95%, Astern 60%
	No. of Locks	6
SECTION II	Iroquios Locks to Port Colborne through Welland Canal	
	Total Time Taken	32 hours 45 minutes
	Total Movements	211
	Power Variation % During Manoeuvring	Ahead 90%, Astern 60%
	No. of Locks	8

N.B. Assuming an average of 400 movements per trip, i.e. 800 movements per return voyage of which there are around 20 per season. Then number of movements per season is in excess of 16,000, very high compared with average number of movements of sea going vessels which does not exceed 5,000 per year.

TABLE 8.3

EXAMPLE OF TYPICAL VOYAGES IN THE GREAT LAKES AND THE SEAWAY

	Distance		Typical Bulker Investigated by Mirrlees National		HALCO Bulker Loaded From		HALCO Bulker Light From	
	Statute Miles	Nautical Miles	Montreal to Time Hrs.	Thunder Bay $ of shp	Duluth to Time Hrs.	Halifax $ of shp	Montreal to Time Hrs.	Duluth $ of shp
Thunder Bay (Duluth)	256 (394)		14.25 -	CSR -	24 4	CSR 20 - 70	24 2	CSR 30 - 60
Gros Cap	62		6.5	25	5	20 - CSR	5	20 - 60
Detour	224		12.5	CSR	12.7 2	CSR 50 - 60	15 -	CSR -
Sarnia	87		8.25	25	9	30 - 75	8	50 - 60
Detroit River	219		12.5 -	CSR -	12 3	CSR 20 - 75	14 1	CSR 20 - 80
Port Colborne	27		15.0	8	11	0 - 30	11	0 - 20
Port Weller	160		8.9 -	CSR -	11 -	CSR -	11 -	CSR -
Cape Vincent	184		24.7 -	20 -	20 -	0 - 80 -	9 10	10 - 20 20 - 80
Montreal	-	140	- -	- -			- -	- -
Quebec	-	269	-	-	1 72	10 - 50 CSR	-	-
Port Cartier	-	609	-	-	-	-	-	-
Halifax								

* CSR - Continuous Service Rating = 85$ MCR - Maximum Continuous Rating

TABLE 8.4

OPERATING PROFILE OF A TYPICAL GREAT LAKES BULK CARRIER

ANNUAL STATISTICS	
Operating Season	259 days
No. of Voyages (round)	20
Length of Voyages	13 days
Lay Time	106 days

UTILIZATION OF THE MAIN ENGINE PER TRIP	Hours	% Time
Full Speed (Continuous Service Rating - CSR)	134	56
Canalling (0 - 30% MCR)	38	16
Confined Waters (30 - 60% MCR)	66	28
Total Time of Main Engine Operation	238	100

TABLE 8.5

ASSUMPTIONS TO CALCULATE AMOUNT OF LOST MONEY BY UNECONOMICAL OPERATION OF CANADIAN LAKERS DURING ONE YEAR

STEAMERS	57 ships
Average Power	4476 kW
Specific Fuel Consumption at MCR	300 g/kWh
Specific Fuel Consumption at 45% MCR	450 g/kWh
Operating Time at MCR	2680 hours
Operating Time at 45% MCR	2080 hours
MOTORSHIPS	58 ships
Average Power	5370 kW
Specific Fuel Consumption at MCR*	195 g/kWh
Specific Fuel Consumption at 45% MCR*	205 g/kWh
Specific Lubrication Oil Consumption	2 g/kWh
PRICE OF FUEL AND LUBRICATION	
IFO 180 cSt	U.S. $ 175 per tonne
MDO	U.S. $ 250 per tonne
Lub Oil	U.S. $ 480 per tonne
MAINTENANCE AND REPAIR COSTS	
Maintenance and repair cost to fuel cost ratio	10 %
Increase of the maintenance and repair cost due to:	
a) Changing steam turbine to diesel engine	50 %
b) Changing main engine fuel from diesel oil to IF 180 cSt	20 %

* Including auxiliary diesel engines fuel consumptions assuming their power = 8% of the main engine MCR.

TABLE 8.6

CALCULATION OF THE AMOUNT OF LOST MONEY, IN US$, DUE TO THE UNECONOMICAL OPERATION OF CANADIAN LAKERS DURING ONE YEAR

STAGE	CALCULATIONS		RESULTS
	A) Use of Steam Propulsion instead of Diesel Engines		
1.	Fuel consumed by steamers per year:		
	0.3 kg x 4476 kW x 2680 hours x 57 ships		205,126 t
	0.45 kg x 4476 kW x 2080 hours x 57 ships		238,804 t
		Total	443,930 t
2.	Fuel which would have been consumed by motor ships of the same power:		
	0.195 kg x 4476 kW x 2680 hours x 57 ships		133,332 t
	0.205 kg x 4476 kW x 2080 hours x 57 ships		108,788 t
		Total	242,120 t
3.	Savings in the fuel consumption: 443,930 t - 242,120 t		201,810 t
4.	Cost of saved fuel (IF 180 cSt): 201,810 t x $ 175	Approx.	$35,371,000
5.	Cost of diesel engine lubricating oil per year: 0.002 kg x 4476 kW x 4760 hours x 57 ships x $0.48 per kg	Approx.	$ 1,166,000
6.	Cost of increased maintenance: 50% x (10% x fuel cost) = 50% x 242,120 t x $ 175	Approx.	$ 2,119,000
7.	Net cost of saved fuel A) = 4 - 5 - 6 $35,371,000 - $1,116,000 - $2,119,000		$32,032,000
	B) Burning MDO instead of IFO 180 cSt in Motor Ship Engines		
8.	Fuel Consumed by motor ships per year*:		
	0.18 kg x 5370 kW x 2680 hours x 58 ships		130,248 t
	0.19 kg x 5370 kW x 2080 hours x 58 ships		123,089 t
		Total	273,337 t
9.	Average difference in prices of MDO and IFO 180 cSt in U.S. $: $ 250 - $175		$ 75
10.	Saving per year: 273,227 t x $75	Approx.	$20,500,000
11.	Increase in maintenance cost: 20% x 10% x 272,337 t x $175	Approx.	$ 957,000
12.	Net savings B) = 10 - 11 $20,500,000 - $957,000		$19,543,000
	C) Total Possible Savings Per Year Which Are Actually Lost = 7 + 12 $32,032,000 + $19,543,000		$51,575,000

* Without auxiliary engines.

Figures 8.7 to 8.9 show three options of the proposed system. Power distribution between the two main engines, 75/25%, would give the opportunity to operate considerably more than 50% of the time with one engine ("father") only, as it was found that many vessels operate at the continuous service rating of 70 to 80% MCR.

The smaller engine with 25% MCR would be additional power for operation at sea, in ice, and during manoeuvres. Due to the substantial difference in power, the three diesel engines ("father", "son" and generator engine) have different sizes of cylinder. This inconvenience would be offset by the advantage of employing the fewest number of cylinders in each engine, improving efficiency and decreasing the maintenance requirement. The ATLANTIC SUPERIOR is fitted with three auxiliary engines each of 900 kW. The proposed system eliminates two of these auxiliary engines. Their function would be taken by the smaller main engine and the turbo-generator or PTO.

The three options differ only in the extent of sophistication. Option 3 would allow feed back to the propulsion system of the surplus turbine output which can occur when developing continuous service power. This system will supply the surplus electrical power generated from the waste heat by the use of the PTO generator fitted to the shaft as a motor. This system called "SSG", has recently been developed in Japan. The other feature of this system is more important for lakers. In the case of low load operation, when recovered energy is not high enough, then the turbo-generator would be backed up by the main engine to meet the electrical energy requirement. This system is already in operation on a ship of the Neptune Orient Line being equipped with one 10 PC4 Pielstick engine. For this ship the calculated pay back time of the extra equipment, in the worst case, is less than 3 years.

The efficiency of the proposed installation could also be improved in the future by replacing the steam turbine with the more efficient organic fluid turbine, provided the development of this engine will meet marine requirements.

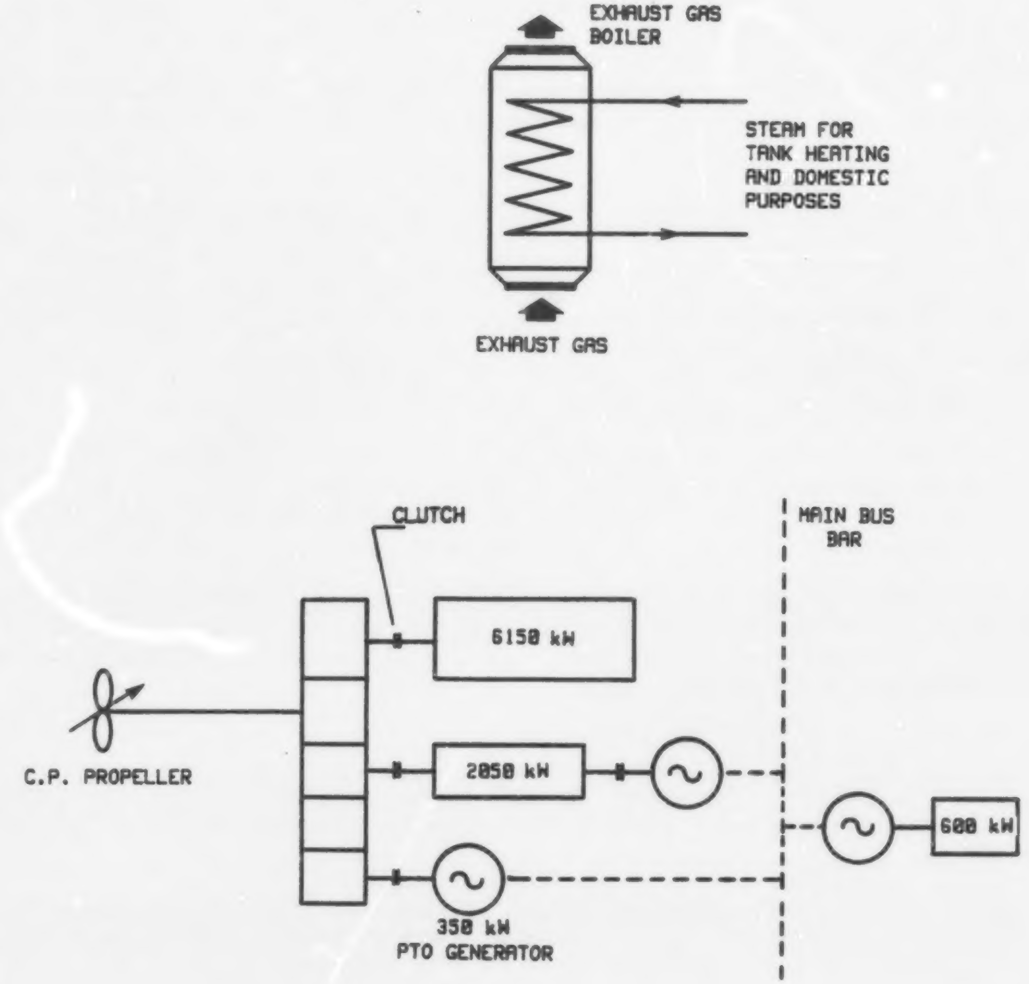
For most vessels about 6-8% of the main propulsion engine output is required to service the power requirements of the electrical auxiliaries and other ship service duties. Therefore, for an engine rated at 6150 kW in the case of Figures 8.7 to 8.9, the electrical power requirement would be between 370-500 kW.

In order to assure the greatest margin between electrical power demand and the power developed by turbo-generator, and to reach the highest fuel savings, the following features should be also applied:

- Most highly effective steam generation by waste heat boiler, e.g., dual pressure.
- Utilization of the heat contained in the cooling water and charge air for space and feed water heating.
- Uncooled turbo-chargers and exhaust valves (to provide more energy to the waste heat boiler).

FIGURE 8.7

GREAT LAKES BULK CARRIER PROPULSION SYSTEM
OPTION No 1


EXHAUST GAS
BOILER
STEAM FOR
TANK HEATING
AND DOMESTIC
PURPOSES
EXHAUST GAS
CLUTCH
MAIN BUS
BAR
6150 kW
C.P. PROPELLER
2050 kW
600 kW
350 kW
PTO GENERATOR

MELVILLE SHIPPING LTD.

FIGURE 8.8

GREAT LAKES BULK CARRIER PROPULSION SYSTEM
OPTION No 2

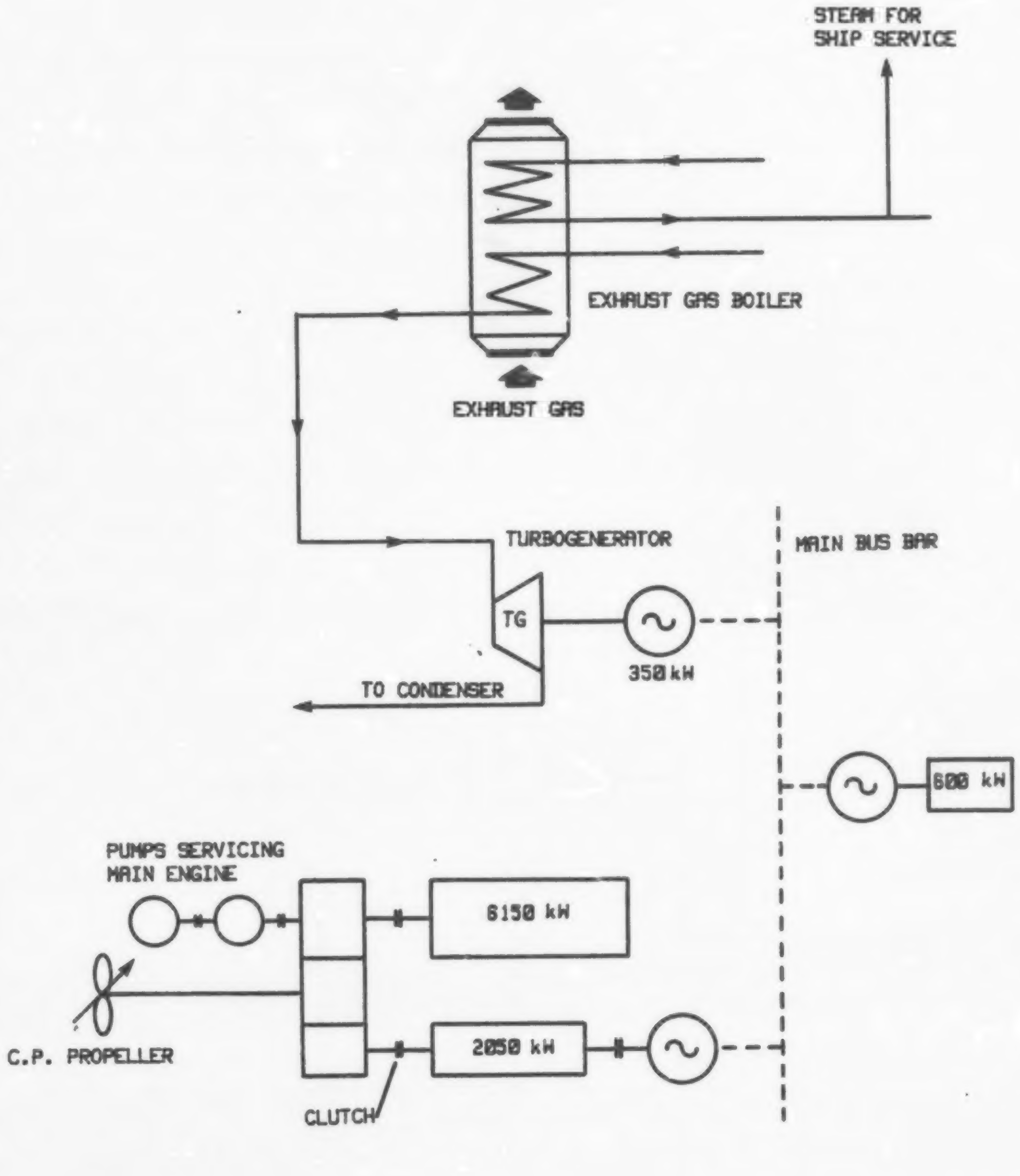
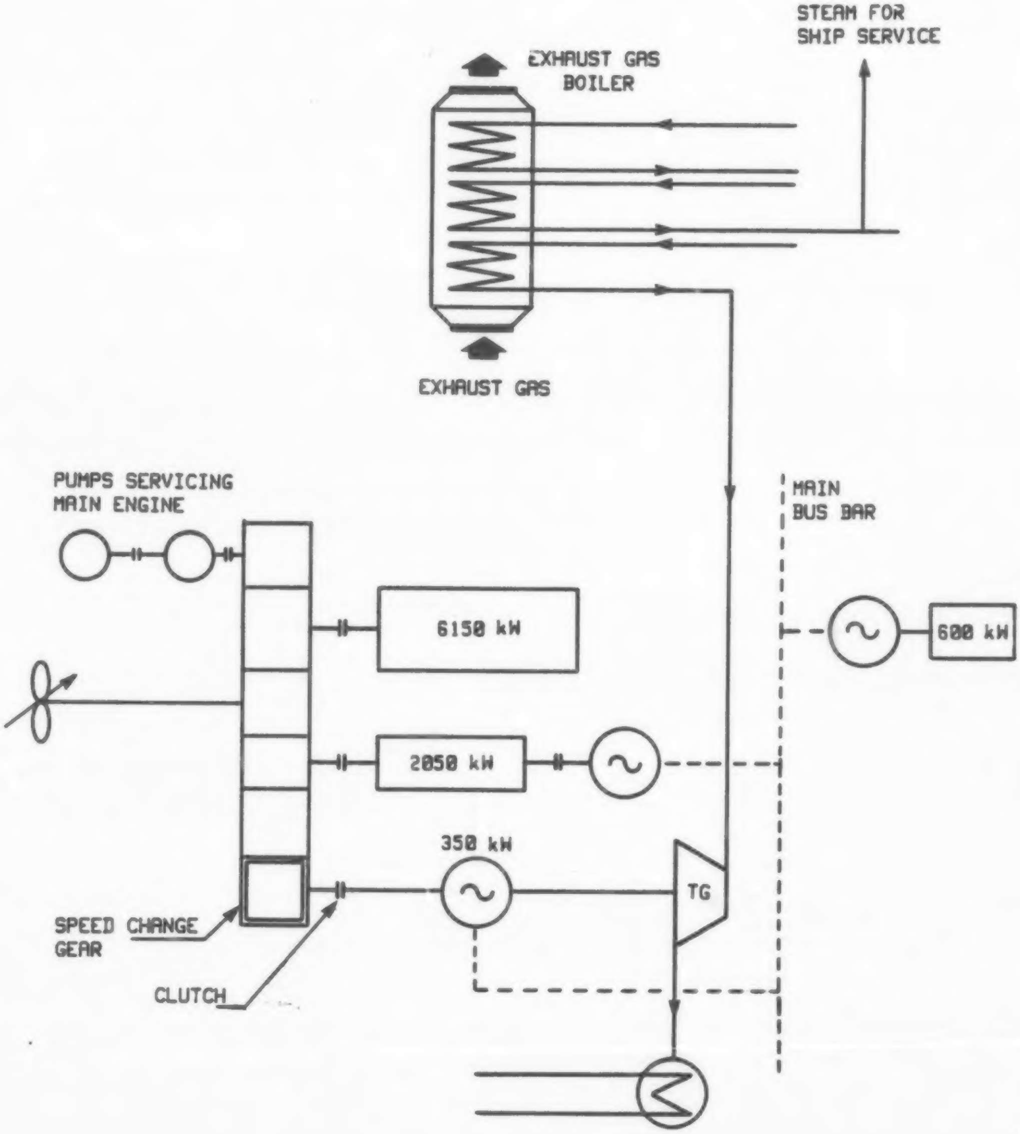

STEAM FOR
SHIP SERVICE
EXHAUST GAS BOILER
EXHAUST GAS
TURBOGENERATOR
MAIN BUS BAR
TG
350 kW
TO CONDENSER
600 kW
PUMPS SERVICING
MAIN ENGINE
6150 kW
C.P. PROPELLER
2050 kW
CLUTCH

FIGURE 8.9

GREAT LAKES BULK CARRIER PROPULSION SYSTEM OPTION No 3


STEAM FOR SHIP SERVICE
EXHAUST GAS BOILER
EXHAUST GAS
PUMPS SERVICING MAIN ENGINE
MAIN BUS BAR
6150 kW
600 kW
2050 kW
350 kW
TG
SPEED CHANGE GEAR
CLUTCH

- All pumps to service the main engine and steam system driven by PTO from main engine to reduce the electrical power demand while at sea. This is easier to design for a geared medium speed engine system than for other propulsion systems.

Finally, the most important characteristic of the three options, which have been proposed, is the ability of all engines to burn the heaviest grades of residual fuel oil, thus creating a so called "one fuel ship".

The operating modes of the proposed systems would be as follows:

a) At Sea
One or two main engines in operation. Electrical power demand satisfied by a shaft generator or turbo-generator depending on chosen option.

b) During Manoeuvres
Electrical power to be supplied by the auxiliary diesel generator and the generator clutched to the smaller main engine which simultaneously could operate for propulsion purposes. In Option 1, the CP propeller would permit the shaft generator to be used to supply electrical power.

c) During Loading Operation in Port
The 600 kW diesel generator would be required to supply only sufficient energy for the ballast operation.

d) During Unloading Operation in Port
The 2050 kW alternator powered by the smaller main engine would give enough power to satisfy the demand of the unloading gear and the ballast system.

The economical use of all diesel engines is the principal advantage of the system. The capital cost would not be as high as a slow speed diesel engine plant since:

- the medium speed diesel engine propulsion plant acquisition cost is usually lower than slow speed diesel engine plant

- elimination of two auxiliary diesel engines will result in a substantial saving

A lower cost for maintenance and repairs would also be expected due to the long layout periods of the two smaller diesel engines during single operation of the main engine.

The cost of a slow speed diesel engine versus two medium speed engines has not been considered here as this has already been discussed in Section 3.2. The aspect of difference in weight between the slow and medium speed engine options should be emphasised, because the proposed systems would be much lighter than the slow speed engine system. This feature could permit further benefits due to a larger cargo capacity.

Estimation of the principal costs and benefits based on 1983 prices, have been assessed and shown in Table. 8.7.

As a further illustration, Figure 8.10 shows the overall cost of generating electrical power by the use of several systems. The figure clearly shows the influence of the "free" electrical power generated by the waste heat driven turbo-generator unit. This influence is strongly conditioned by the number of days at sea when the waste heat boiler can generate sufficient steam to satisfy the electrical load. The figures are derived by MAN/B&W by means of a computer based "tool" able to give an economic evaluation of alternative generator installations. The calculation, made in 1983, was based on the following main assumptions.

- Although three individual generators are necessary, only one set is normally running at sea as follows:
  - Case 1 - a diesel generator
  - Case 2 - a main engine driven generator
  - Case 3 - a turbo-generator.
- Generator power is equal to 625 kW (all generators).
- All plants are assumed to be financed with 80% of the installation costs by means of series loans over 9 years at an interest at 12% p.a. and top-up financing at an interest rate of 8% p.a. with a pay back period of 3 years.
- The power profile used:

manoeuvring	-	400 hours/year at 700 kW
ballast	-	3,500 hours/year at 500 kW
loaded	-	3,500 hours/year at 400 kW
port	-	1,360 hours/year at 330 kW

While these assumptions are not representative for lakers, this form of analysis is useful to allow the best selection of generating equipment to be made for all ships.

The question of why many shipowners do not utilize, even in newbuildings, the advantages of recently developed fuel saving measures is difficult to answer. The most probable reason is that conservative thinking within the shipping industry is just as applicable to the Canadian as the foreign owner. The strong feeling by shipowners that equipment provided in ships must be of proven reliability is clearly evidenced, and many owners are reluctant to specify machinery which has enjoyed less than a five year proving period at another owner's expense. This type of conservatism is not so prevalent in other industries which are much more receptive to the introduction of new ideas and equipment.

8.4.4 Energy Efficiency Measures for Existing Vessels by Retrofitting

At the present time the largest potential for energy saving in the Great Lakes fleet is by improvements to existing vessels. Many of the vessels are of considerable age and powered by steam turbine plant of vintage design. There is a particular need to improve the performance of these vessels to remain competitive with more modern diesel engine powered ship.

TABLE 8.7

COSTS AND BENEFITS OF THE PROPOSED PROPULSION SYSTEMS

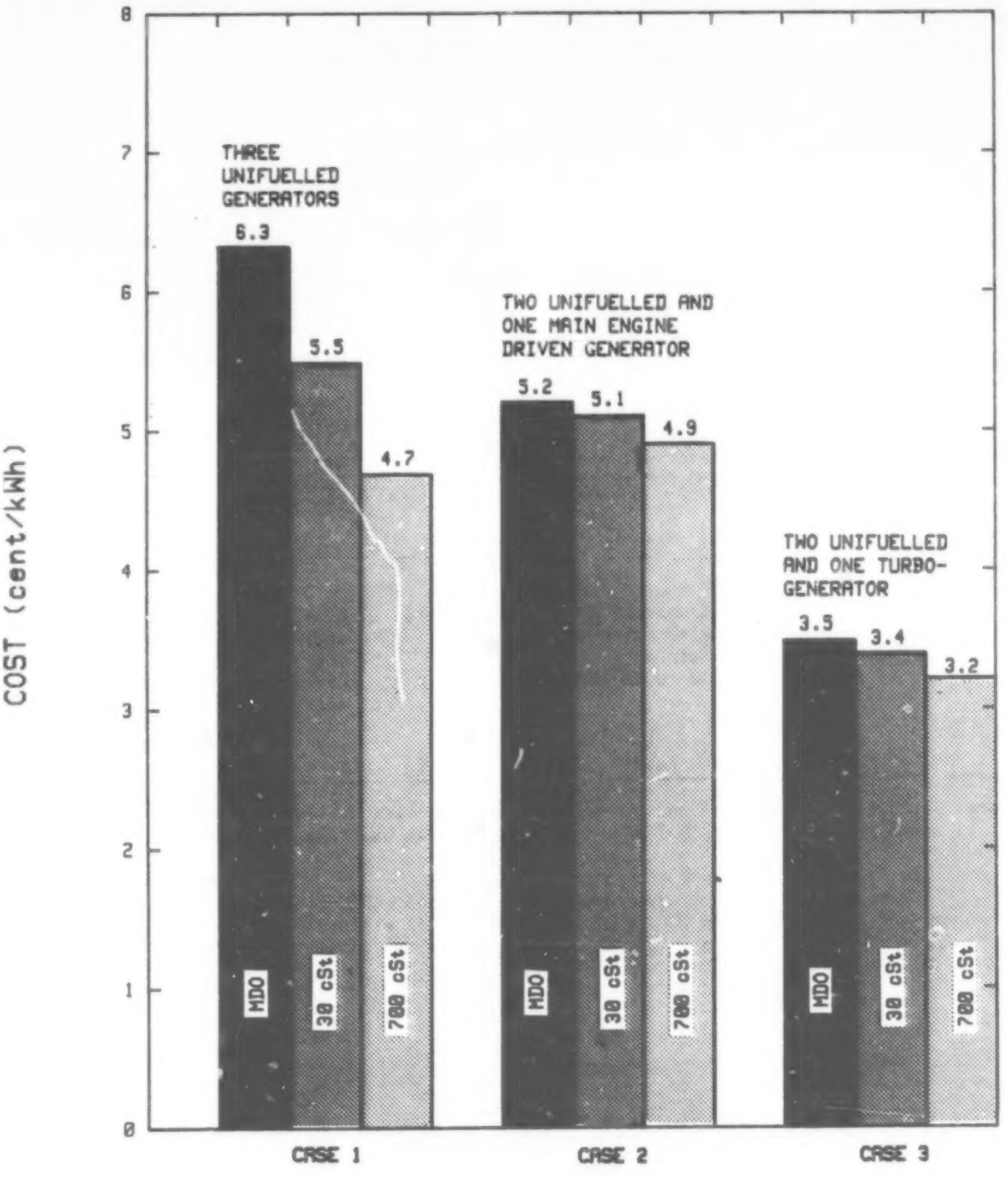
OPTION	COSTS	BENEFITS
Option No. 1		
Main engine driven shaft generator.	Capital cost of a shaft generator with gearing, additional equipment, materials and installation Can $130,000 to 190,000 approx.	a) Elimination of two auxiliary diesel engines of $320,000 each. Total, two engines yield saving Can $640,000 approx. b) Fuel savings in the range of Can $10,000 per year.
Option No. 2		
Turbo-alternator	Capital cost of a turbo-alternator set of 350 kW equal to Can $350,000 to 450,000 approx.	a) Elimination of two auxiliary diesel engines, saving Can $640,000 approx. b) Fuel savings in the range of Can $68,000 per year.
Option No. 3		
Main engine driven shaft generator and turbo-alternator.	Capital cost of a shaft generator, turbo-alternator and constant speed gearing equal to Can $500,000 to 800,000 approx.	a) Elimination of two auxiliary diesel engines, saving Can $640,000. b) Fuel saving in the range of Can $70,000 per year. c) Additional savings due to possibility of electrical power feed back to the propulsion system.

Note: Calculations of cost and savings have been made as shown in Table 8.6. The assumptions from Table 8.5 and the following have been used in the calculations:

2680 hours/year of shaft generator work at CSR
1320 hours/year of shaft generator work at 30-60% MCR
30 g/kWhr - difference in sfc of main and auxiliary diesel engine at CSR
20 g/kWhr - difference in sfc of main and auxiliary diesel engine at 30-60 % MCR

FIGURE 8.10

GENERATION COST FOR ALTERNATIVE SYSTEMS



[43]

SPECIFIC COST IS STATED FOR THREE DIFFERENT FUEL OILS;
MDO, 30 cSt AND HEAVY FUEL 700 cSt (50°C)

MELVILLE SHIPPING LTD

There are 38 lakers powered by steam turbines and some of these ships are relatively new when compared with the average age of the Great Lakes fleets. Twenty-three steam ships were built after the year 1958 and 9 of them after 1963.

The newer vessels are best suited for replacement of their propulsion machinery since their remaining life time is expected to be very long. Two alternatives should be considered:

- Re-engining by replacement of the steam turbine by diesel engine.
- Upgrading of the existing steam cycle and associated components and derating of the main propulsion turbines and gears.

Given the high cost of energy; the investment in the ship and the need for the vessel to remain competitive; the option to re-engine or upgrade the existing system must be carefully evaluated. A simple pay back period analysis is usually not sufficient. A more sophisticated approach is required which takes into account the numerous parameters, including the future growth of the shipping market. This study performs only a general evaluation of these alternatives to assess the possible advantages or disadvantages.

a) Steam to Diesel Engine Conversion

The assessment of savings is partially supported by figures taken from the report on modernization of the US domestic bulk fleet issued in 1983 under sponsorship of US Department of Transportation [105]. Based upon sound technical and economic evaluation criteria, an economic analysis of alternatives was made for replacing existing inefficient steam turbo-electric plant with a diesel engine system. This option applies to approximately 85 U.S. vessels of the T-2 type of 25,000 dwt ships built during World War II and fitted with 5000 kW steam turbo-electric propulsion plants. The size of the ship and type of operation does not differ significantly from Canadian lakers since the vessels are used on short sea routes and coastal service.

The study revealed that all re-engining alternatives were considered viable and re-engining to slow speed diesel as well as to medium speed had acceptable returns on investment. These results could serve as guidance to Canadian shipowners willing to modernize their fleets.

An analysis of the profitability of re-engining Great Lakes ships to improve the fuel consumption, based upon the assumptions given in Table 8.8, shows the net present value (NPV) after 10 years as minus Can $3,800,000. The negative NPV indicates, the cost of the conversion would not be justifiable over a 10 year pay back period. Therefore replacing the steam turbines of Great Lakes bulk carriers with diesel engines is not recommended.

However, in some cases re-engining may be justified. It would depend on actual fuel consumption of steam turbine plant, actual shipyards' costs, possibilities of sale of existing machinery, etc. An accurate economic analysis would be necessary in each individual case, and it could reveal quite different figures, which may be acceptable to the shipowner.

b) Upgrading the Existing Steam System

Modifications of the existing propulsion plant to save fuel requires a smaller initial investment than for full re-engining to diesel, but savings are also smaller. The assessment of possible fuel savings is based on investigations carried out by General Electric and Stal-Laval as well as on published results of trials of modified ships.

On a Canadian laker the possible percentage of fuel saving at idealized conditions are as follows:

- Replacement of the propeller with one designed for slow steaming resulting in an improvement in propulsive efficiency. The changed propeller will usually have the largest possible diameter acceptable by the stern frame and should operate at the slowest possible speed.	7.5%
- Replacement of the high pressure turbine first reduction gears and pinion to return H.P. turbine speed close to its original r.p.m.	2%
- Recovery of H.P. turbine efficiency by renozzling the H.P. turbine first stage nozzle plate and re-diaphragming (partial arcing) of the stationary stages after the first stage.	2.5%
- Addition of a cascade bleed system and shifting of the existing L.P. turbine extraction point to a bleed belt higher up in the L.P. turbine steam path to improve L.P. heater performance.	1.4%
- Modification to boiler burners, superheaters, furnaces, air heaters, feed pumps, excess air monitoring, etc.	2.5%
Total Saving	15.9%

Using the assumptions made earlier (Table 8.8) to estimate the benefits of the steam to diesel conversion and by means of data obtained from manufacturers, the following figures for upgrading the steam cycle are derived:

- The acquisition and shipyard cost Can $	1,800,000
- Fuel saving per year (tons)	1,175
- Cost of saved fuel for first year Can $	264,400
- Net present value over 10 years Can $	603,400
- Payback period (years)	8

The profitability of an investment in improvements to reduce the fuel consumption of lakers is heavily dependant upon the length of time used to write down the initial capital outlay. Although 10 years is commonly used to recover the investment, in the case of lake ships which enjoy an extended life span, there could be a strong argument to prolong this period.

TABLE 8.8

ASSUMPTIONS FOR ECONOMIC EVALUATION OF CANADIAN LAKERS RE-ENGINING (STEAM TO DIESEL)

Propulsion Power (MCR)	6700 kW
Ship's Operating Time	
. at CSR = 85% MCR	2680 hours
. at 45% MCR	2080 hours
Average sfc of a Steam Turbine Plant	
. at CSR = 85% MCR	300 g/kWh
. at 45% MCR	450 g/kWh
Average sfc of a Diesel Engine Plant (together with auxiliary diesels)	
. at CSR = 85% MCR	190 g/kWh
. at 45% MCR	200 g/kWh
. Lubricating oil consumption	2 g/kWh
Costs	
. Acquisition cost of the main diesel engine of 6700 kW approx.	Can $2,700,000
. Acquisition cost of three auxiliary diesel generators	Can $1,000,000
. Estimated shipyard costs	Can $9,700,000
. IFO 180 cSt price	Can $225 per tonne
. MDO price	Can $325 per tonne
. Lub oil price	Can $625 per tonne
. Maintenance and repair cost to fuel cost ratio	
- for diesel engine	10 %
- for steam turbine	6 %
. Annual escalation in fuel costs	10 %
Financing	
. Loan term	10 years
. Interest rate	10 %
Savings in Fuel Costs (Calculation based on above stated assumptions)	Can $649,900 for first year

Note: Since the acquisition costs of a slow speed diesel engine is not substantially higher than of a medium speed engine with gearing, the assumed cost can roughly be applicable to both types of engines.

c) Other Energy Efficiency Measures

Vessels which are presently powered by diesel engines may be able, after appropriate conversions are made, to burn cheaper but heavier grades of fuel. Unfortunately not all engines are suitable for conversion to burning the heavy residual fuel oil or the manufacturer does not market conversion kits. If the engines can be modified to burn heavy fuel, the fuel oil system must be modified as well as the engine. Engine modification would include some or all of the following:

- New fuel valves and pumps
- Modified inlet and exhaust valves
- Modified cam shaft
- New pistons
- Modification to the turbo-charger
- Modification to or change of the cylinder heads
- Changes to cooling system

Modification to the fuel system would require the following:

- Installation of heating coils in the heavy fuel tanks
- Fuel oil purifier system and heaters
- Fuel lines to be lagged and sometimes trace heating to be fitted
- Installation of new fuel filters
- Installation of fuel booster pumps and fuel circulation system
- Installation of fuel heater

The conversion costs can often be offset quickly by a reduction of fuel cost.

Although some engines are not capable of burning residual fuel, they can be made to burn lower grade fuel by the use of a fuel blender. In the case of blended fuel not all the modifications listed are required, the actual modification will depend to some extent upon the maximum viscosity of the blend to be burned.

8.5 Ferries

The Canadian coast line has numerous islands and gives rise to the need for ferries to service passenger traffic. More than 50 ferries of 1000 GRT and above are presently in operation, most of which are used to service passenger requirements within the Atlantic provinces and between Vancouver and Vancouver Island.

Some of the ferry routes pose a particular problem due to ice in winter. The Prince Edward Island service, for example, in the shallow water of the Northumberland Strait encounters a barrier created by grounded and ridged ice. This ice can be present from January through to May with ice up to 70 cm thick in March. For this reason a certain number of ferries operating on the Atlantic coast are laid up during the winter time. However, there are some which operate through the winter, among which is the M.V. ABEGWEIT.

This vessel is the most modern ferry in Canada and has been designed for the east coast conditions with fuel economy provisions. The rest of the Canadian ferries present various types of development and have an average age of approximately 19 years.

The life cycle for ferries is estimated by shipowners as 20 years, at least, and it is anticipated that some of the ferries presently in operation will be replaced in the near future. The vessels which are to replace the existing tonnage will be in service up to year 2000. The design of the new vessels should take into account the eventual shortage of light fuels, a deterioration of the quality of heavy fuels as well as an increase in fuel prices. Up to now shipowners efforts in energy saving have been directed to high utilization of the vessels rather than energy efficiency measures.

A generalised approach to the evaluation for the potential to save fuel on ferries cannot be taken due to the many different operating profiles. However, for the purpose of this study two of the most common types have been distinguished:

- short route ferry, often called "sea bus", with trip duration up to one hour
- long route ferry with trip duration more than six hours.

Ferries with crossing times between one and six hours may share characteristic features of the both types.

8.5.1 Energy Efficiency Measures for New Vessels

There are many alternatives to improve the energy utilization open to the designers of ferries. Some of these alternatives can be suitable for use on all types of ferries while others are suitable only for those vessels with longer sea routes. The most significant is the choice of main engine. It is expected that medium speed diesel engines will remain the preferred type of ferry propulsion for the foreseeable future. Their well known advantages for ferry application are:

- low height, which permits through decks and maximizes cargo and passenger accommodation

- lower weight compared with slow speed installations. A weight saving which may be as high as 400 - 600 tonnes is significant when compared with the weight of cargo, 1500 to 2000 tonnes for this kind of ship.

Compared with heavier types of machinery, having the same specific fuel consumption, and for the same ship capacity, the above advantages translate into an overall fuel economy per nautical mile, which may be as great as 4 to 5%, as claimed by S.E.M.T. Pielstick.

The present trend is to select 4 medium speed main engines driving two CP propellers through 2 single reduction gearboxes. This permits good flexibility of propulsion power with the optimum fuel consumption in most operating conditions. In the high season 4 engines are used, while in the low season cruising times are extended and only 2 engines are employed. The Canadian ferry ABEGWEIT [106] is powered by 6 medium speed engines, all of which are run only when the vessel is navigating in ice.

The diesel engines should have the capability to burn the heaviest grades of fuel oil in order to achieve the largest savings. That is why some ferry builders and owners are giving increasing attention to slow speed, two stroke diesel engines. These engines have not traditionally been the first choice for such specialized ships as ferries. The superiority of these types of engines for reliable performance, especially when burning low grade fuel oils, and the oil price boom has resulted in their application to some types of ferries. Only the smallest types of slow speed engines have been fitted to enable them to be fitted below decks. It is of interest to note that recently built cruise ships, which have many similar characteristics of passenger ferries, have been fitted with slow speed oil engines.

Figures 8.11 and 8.12, taken from B & W specifications, show the possible application of B & W slow speed engines. The height for overhauling of pistons has been reduced to a minimum due to a special jib crane design patented by B & W. Since the introduction of the smaller types of slow speed engine, the number of orders for these engines for installation in ferries has accelerated strongly. Canadian ferry operators have not yet installed this type of engine in their ships.

Ferries powered by medium speed diesel engines and CP propellers are especially suitable for PTO installations. Most of the recently built ferries for service across the English Channel, where distances are relatively short, successfully use shaft generators. There is no reason why Canadian ferries could not successfully apply the same practice, and first signs can already be seen. The CN Marine ferry GULF SPAN presently under construction, will be fitted with 2000 kW shaft driven generators, one to each gearbox. The recently built M.V. ABEGWEIT has provisions for power take-off from the gearboxes.

The problem of bunker storage is important because of low ambient temperatures during the larger part of Canadian ferries' operating season. Maximizing efforts in order to decrease heat losses from bunker tanks may give substantial fuel savings during cold weather.

As regard to other energy efficient designs, these would be more suitable only for long route ferries. Particularly, the following measures could be useful:

- Waste heat recovery boiler, even if it is operated only 3 or 4 hours on full load during each trip, would be justified.

- Utilization of waste heat from main engine cooling mediums may be possible in many cases.

- Main engine driven pumps and compressors.

- Main engine and fuel monitoring may be very advantageous due to the high powers applied to ferries. Even small correction of combustion parameters would reduce the fuel bill.

FIGURE 8.11

APPLICATION OF A SLOW SPEED DIESEL ENGINE IN A FERRY

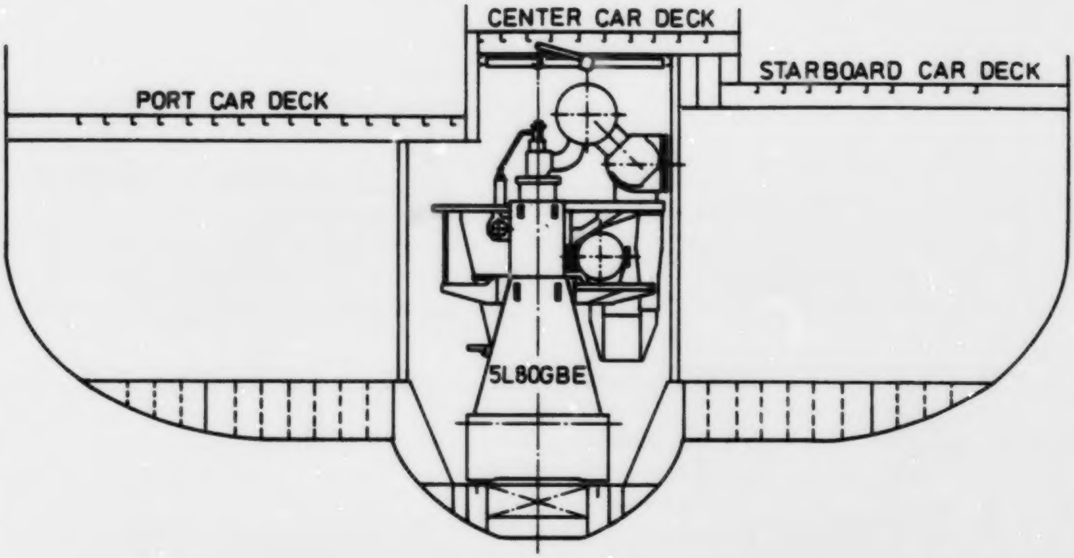
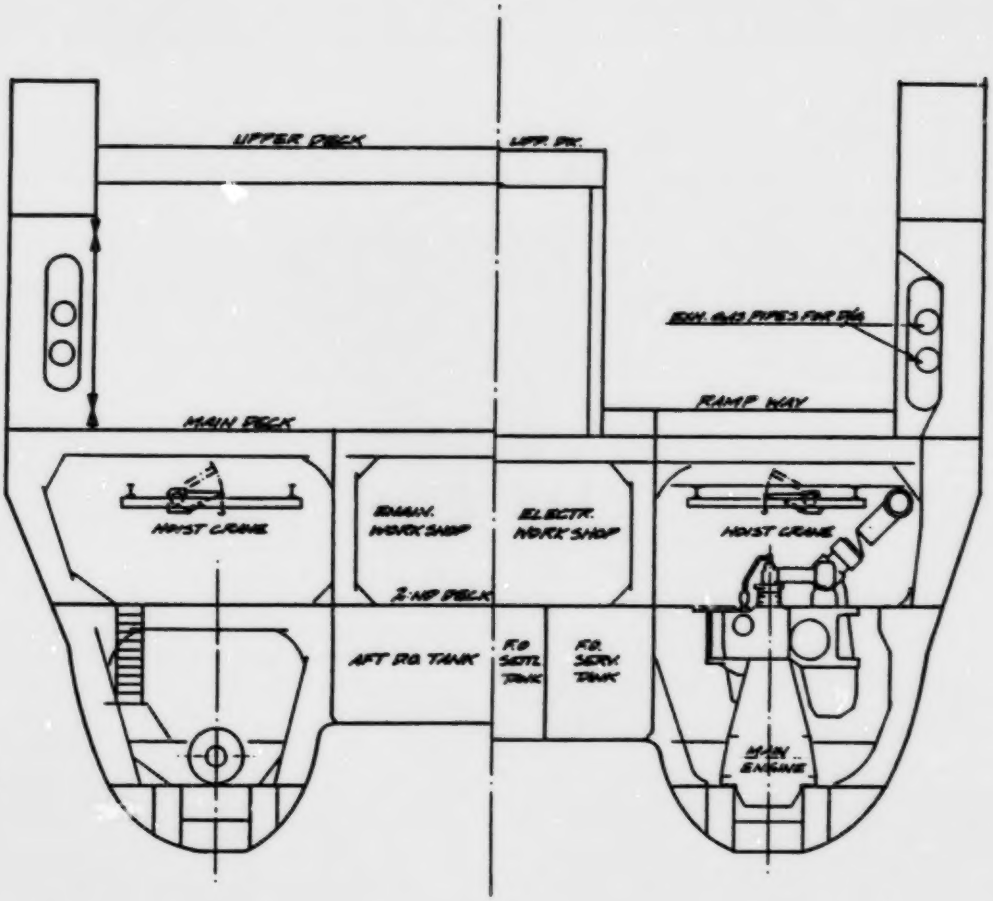

CENTER CAR DECK
STARBOARD CAR DECK
PORT CAR DECK
5L80GBE

FIGURE 8.12

APPLICATION OF A SLOW SPEED DIESEL ENGINE IN A FERRY


UPPER DECK
UPP. DK.
EXH. GAS PIPES FOR D/G
RAMP WAY
MAIN DECK
HOIST CRANE
ENGIN. WORK SHOP
ELECTR. WORK SHOP
HOIST CRANE
2:ND DECK
AFT F.O. TANK
F.O. SETTL. TANK
F.O. SERV. TANK
MAIN ENGINE

- Draft and trim monitoring can also increase fuel efficiency to a greater extent than in cargo vessels, due to of the influence of higher ship's speed.

The use of CNG (Compressed Natural Gas) as fuel for the diesel propulsion engines is presently under active consideration by some ferry companies. It is understood that ten ferries in Vancouver are to be retrofitted with this system. Due to storage problems with the gas it is probable that this propulsion fuel is only suitable for ferries fitted with relatively low powered engines operating on short sea routes. While gas will decrease the fuel bill due to lower price, gas fueled engines are less fuel efficient than diesel fuel engines.

Ferries built in Europe recently are more advanced, in regard to energy conservation, than any built or being built in Canada. The following description of two vessels may serve as an example for designing new Canadian ferries.

- KRONPRINSESSAN VICTORIA with four Wartsila Vasa 12 cylinder V32 medium speed engines developing 15600 kW which drive two CP propellers through two sets of reduction gears. A step-up gear drive to 1500 r.p.m. is incorporated in each gearbox to drive the main alternators of 1100 kW each. Wartsila VASA diesel engines were also chosen for auxiliary use and are a four cylinder in-line version of the main engines. The arrangement of the engines and the reduction gears allows operation of any number of engines in any feasible combination. At partial loads some engines may be shut down and the others will run within a range of lower fuel consumption. All engines are designed to operate on 380 cSt fuel [107].

  Energy from the high temperature fresh water cooling system is utilized for accommodation heating. Oil fired boilers for steam production in port are adapted to burn sludge. When at sea, two exhaust gas boilers supply steam.

  Main and auxiliary engine surveillance is by an Autronica protection system. The ferry was built by the Swedish shipyard Gotaverken Arendal and is operated by Swedish shipowners.

- OLAN HOLLANDIA owned by Olan Line and operated on the English Channel route. This ship is equipped with four in-line 8PC2-5 S.E.M.T. Pielstick engines which burn 180 cSt fuel oil, continuously, including manoeuvring. Flexibility brought about by the ability to run engines as required have resulted in fuel savings of 10%.

8.5.2 Energy Efficiency Measures for Existing Vessels by Retrofitting

As for any other ships, the possibilities of retrofitting of more economical machinery and systems to existing ferries are limited to those modifications which do not need high capital cost nor require large alterations to the vessel's structure.

Typical modifications will be almost the same as for Great Lakes vessels. The most profitable and highly recommended change would be the conversion of machinery to consume heavier grades of fuel oil in main engines as well as in auxiliary diesels and boilers.

Engines which are unable to burn residual fuel may be able to consume blended fuel without extensive modification. In the case of ferries the fuel could be blended ashore in preferrence to the installation of blenders on board. If heavy or blended fuels are used, larger amounts of sludge may be rejected from purifiers. This sludge should be burned in auxiliary boilers and not wasted. The amount of sludge from the purifier system could be sufficient to meet the boiler needs due to the high power of the main engines.

Since the operation profiles of different ferries vary so much it is not possible to provide a general remedy for fuel economy. Each case should be carefully investigated and only then can appropriate measures be chosen. The spectrum of these measures is very wide and not all can find application in Canadian ferries.

8.6 Ocean Going Ships

To be competitive with foreign operators, most Canadian owned sea going ships are registered under foreign flags, and unlike the Great Lakes fleet these ships tend to be of modern design and compatible with good recent international practice.

Canadian deep sea shipowners generally build new ships in Far East shipyards. Those shipyards offer ships which embody recently developed technologies making new vessels efficient and economical. It is a commonly held view that the centre of shipbuilding technology has been transferred from the historic shipbuilding nations of Europe to the Far East, particularly to Japan and Korea. This is especially true for cargo and bulk cargo vessels but not so true for specialised ships.

A study of vessels recently delivered from Japanese shipbuilders shows these vessels to be technically advanced ships. For the past few years the Japanese shipbuilders have concentrated on developing energy efficient ships. This has resulted in the delivery over the last few years of a series of remarkably fuel efficient ships which are attracting worldwide interest. During 1983, in particular, the Japanese R & D activities for energy saving in large vessels have resulted in the introduction of industry standards in several areas of ship design.

Tables 8.9 and 8.10 show examples of energy efficient ships presently under construction or recently delivered from Japanese shipyards [111]. Japanese shipyards also lead in the conversion of older inefficient vessels by retrofitting more economical machinery. Table 8.11 shows some examples of steam turbine vessels re-engined in Japanese shipyards.

Highly efficient vessels are also built in the shipyards of other countries. For instance, Danish shipbuilding industry claims to construct approximately 50% of all sea going ships fitted with waste heat driven turbo-alternators

TABLE 8.9

EXAMPLES OF ENERGY EFFICIENT SHIPS

Ship/Type	Shipbuilder/Owner	Hull	Machinery	Fuel Savings as Compared with Conventional Ship	Major Features for Saving Energy
KINOKAWA MARU Coal/Ore Carrier	Sumitomo Heavy Industries/ Daiichi-Chuo Kisen	L = 285 m B = 47.5 m D = 24.5 m H = 17.7 m	6RLB 90 (Sumitomo/Sulzer) 15,200 kW - 90 r.p.m. 8.0 m propeller 13.5 knots	45%	. Long stroke, H.F.O., fuel efficient engine. . Stern bulb. . Exhaust gas waste heat utilization. . Very low auxiliary electrical power requirement.
SHOHU MARU Ore/Bulk Carrier	Nippon Kaiji Kyokai/Showa Line	L = 260 m B = 43 m D = 23.8 m H = 17.2 m	14PC4V (NKK/Pielstick) 12,500 kW - 64 r.p.m.	40%	. Main engine derated to 11,900 kW and matched to a 64 rpm CP propeller. . Exhaust gas waste heat utilization. . Shaft generator/motor system (SGM) returns surplus power to the propeller shaft. . Anti-fouling paint. . Improved hull form.
SHINHO MARU Ore/Bulk Carrier	Mitsubishi Heavy Industries/Shinwa Kaiun	L = 302.5 m B = 50 m D = 24.6 m H = 18.3 m	6VEC 60/150 H x 2 (Mitsubishi) 14,100 kW/140/60 r.p.m. 13.0 knots	-	. "Mitsubishi Bow" and special shape of the stern. . 9.3 diameter propeller. . Two slow speed diesels connected via clutches and a reduction gear to a 57 rpm propeller. . Mitsubishi Electronic Diesel Engine Aids (MEDEA). . Waste heat recovery turbo-generator plant and thyristor-controlled compact shaft generator acts as a motor to drive the propeller shaft. . Drastic reduction of electric power demands. . Mitsubishi reaction fin. . Self-polishing long life anti-fouling paint. . Mitsubishi fuel saving navigation system TONAC Pilot.
HUEI MARU Ore/Bulk Carrier	Kawasaki Heavy Industries/Nippo Kisen	L = 350 m B = 50 m D = 24.6 m H = 18.3 m	K8SZ70/150 C (Kawasaki - MAN) 11,500 kW/126/45 r.p.m. 12.4 knots	50%	. Highly improved hull form. . Kawasaki-Escher Wyss CP propeller, 11 m in diameter, at 45 rpm. . Slow speed diesel fitted to a reduction gear. . Waste heat recovery system. . A shaft generator/motor system (SGM) returns surplus power to the propeller shaft.

TABLE 8.10

EXAMPLES OF ENERGY SAVING SHIPS UNDER CONSTRUCTION BY MAJOR JAPANESE YARDS

	Use	Shipbuilder	Hull	Machinery	Propeller Generator	Ship speed Fuel saving rate	Special items for saving energy
A	Ore-Coal carrier	Kawasaki Heavy Ind.	208,000DWT LBDd=305 ×50×24.6× 18.3	Kawasaki-MAN K8SZ70/150C 15,500PS× 126/45rpm	CPP TG540kw×1 DG800kw×1 SG150kw×1	12.5KT 43.5T/day (−39%)	• Improved block coefficient • Reduction gear and flexible coupling • 3 blade propeller • Exhaust gas turbo generator • High tensile steel
B	Bulk carrier	Mitsubishi Heavy Ind.	207,000DWT LBDd=302.5 ×50×24.6× 18.3	Mitsubishi UE 6UEC60/150H×2 18,900PS× 140/60rpm	CPP TG600kw×1 DG600kw×2 SG280kw×1	13.0KT 53T/day (−27%)	• Exhaust gas turbo generating plant • Improved main engine • Low speed propeller • Reaction fin
C	Ore-Coal carrier	Sumitomo Heavy Ind.	177,000DWT LBDd=285 ×47.5×24.5× 17.7	Sumitomo Sulzer 5RLB76 12,900PS× 120/50rpm	CPP TG 1 DG 2	12.0KT 35T/day) (−38%)	• Stern bulb • Low speed propeller • High tensile steel • Self polishing paint • Improved main engine • Effective use of waste heat
D	Ore-Bulk carrier	Nippon Kokan K.K.	141,900DWT LBDd=260 ×43×23.8× 17.2	NKK-Pielstick 14PC4V 16,700PS× /64rpm	CPP TG800kw×1 DG640kw×2 SG/M 100kw/250kw×1	13.8KT 1,900kg/h (−37%)	• Derating engine • Low speed propeller • Effective use of waste heat • Shaft generator and motor system • Improved hull form
E	Oil tanker	Mitsui Engineering & Shipbuilding Co.	66,000DWT LBDd=205 ×35.5×19 ×13	Mitsui BIW 7L67GBE 14,600PS 121/60rpm	FPP TG500kw×1 DG630kw×2 SG120kw×1	15KT 41T/day (−35%)	• Integrated duct propeller • Low speed propeller • Derating engine • Effective use of waste heat
F	Bulk carrier	Hitachi Shipbuilding Ltd.	61,000DWT LBD=215 ×32.2×17.8	Hitachi BdW Twin Bank 2×7L45GTBE 12,300PS /70rpm	CPP TG500kw×1 DG500kw×2 SG500kw×1	14.5KT 35T/day (−27%)	• Twin bank engine • HZ nozzle • Effective use of waste heat • Improved navigation
G	Bulk carrier	Ishikawajima Harima Heavy Ind.	60,850DWT LBDd=215.4 ×32.2×17.8 ×12.4	IHI Pielstick 6PC4-2L 9,540PS× 400/77.9rpm	FPP TG800kw×1 (SG) DG500kw×2	13.5KT 30.7T/day (−33.5%)	• Improved main engine (MPC system) • Effective use of waste heat (mixing pressure type turbo generator)

TABLE 8.11

TYPICAL CONVERSIONS OF LARGE STEAMSHIPS TO DIESEL SHIPS

Ship's name	Shipowner	When built	DWT	Engine Output (before converting) PS	Engine Type	Output PS	Shipbuilder	When converted
Oil Tanker								
Mobil Hawk	Mobil Shipping and Trans. Co.	1976	285, 440	34, 400	12PC4 V×2	36, 000	IHI Aioi	1978Sep-Nov
Alva Sea	Alva Sea Shipping Co.	1973	221, 457	32, 450	16V52/55×2	33, 760	MHI Yokohama	1979Jan-July
Seiko Maru	Sanko Kisen	1979	248, 228	36, 000	18V52/55A×2	36, 000	KHI Sakaide	1979March-
Takasaka Maru	N Y K	1976	280, 618	36,000	12PC4 V×2	36, 000	IHI Aioi	1980Mar-June
Atlantic Venture	Golden Peak Maritime Agencies-Ltd.	1969	133, 429	23, 500	12V52/55A×2	25, 320	MHI Yokohama	1980Mar-June
Mobil Eagle	Mobil Shipping and Trans. Co.	1976	285, 440	36, 515	12PC4 V×2	36, 000	IHI Aioi	1980June-Aug
Tokiwa Maru	N Y K. Okada Shosen.	1973	237, 458	34, 000	16V52/55A×2	33, 760	MHI Nagasaki	1980June-
Amoko Seafarer	Amoco Intern. Oil.	1974	273, 263	36, 000	12L58GFCA×2	35, 800	Mitsui Yura	1980June-Sep
Mobil Falcorn	Mobil Shipping and Trans. Co.	1975	281, 503	38, 000	12PC4 V×2	38, 000	IHI Aioi	1980Aug-Oct
Athos	Mobil Oil Franc.	1975	276, 234	38, 000	12PC4 V×2	36, 000	•	1980Oct-1981
D'artagnan	•	•	•	•	•	•	•	1981Jan-Apr
Valencia	Compania Espanola de Petroleos S. A.	1977	173, 288	32, 000	12V52/55A×2	25, 320	Hitachi Sakai	1981Feb-May
Saudi Glory	Mobil Shipping and Trans. Co.	1974	276, 388	38, 000	12PC4 V×2	36, 000	IHI Aioi	1981Apr-June
Al Haramain	Mobil Shipping and Trans. Co.	1975	281, 596	38, 000	12PC4 V×2	36, 000	•	1981July-Sep
Container Ship								
Kasuga Maru	N Y K	1976	43, 896	80, 000	8 RND90M	55, 600	MHI Kobe	1980Feb-June
Rhine Maru	M O L	1972	35, 544	80, 000	7 K90GFCA	55, 200	Mitsui Tamano	1980July-Nov
Kitano Maru	N Y K	1972	35, 198	80, 000	8 RND90M	53, 600	MHI Kobe	1980Aug-Dec
City of Edinburgh	Bernard Street Holdings Ltd.	1973	49, 590	88, 000	K7SZ80/160BL	51, 380	KHI Kobe	1980Oct-1981Mar
Candigan Bay	Overseas Containers Ltd.	1973	48, 554	81, 120	8 RND90M	50, 880	MHI Kobe	1980Dec-1981
Tokyo Bay	•	•	•	•	•	•	•	• May
Kerrigan	Compagnie Generale Maritime.	•	49, 690	88, 000	•	53, 600	MHI Yokohama	1981Jan-May
Benalder	The Ben Line Steamers Domina Container Ship.	1972	49, 590	88, 000	K7SZ80/160BL	51, 380	KHI Kobe	1981Apr-Aug
Liverpool Bay	Overseas Containers Ltd.	1973	48, 554	81, 120	8 RND90M	50, 880	MHI Kobe	1981May-Sep
Osaka Bay	•		48, 542	•	•	•	IHI Aioi	1981June-Oct
Benavon	The Ben Line Steamers Domina Container Ship.	1973	49, 590	88, 000	K7SZ80/160BL	51, 380	KHI Kobe	1981Aug-1982
Kowloon Bay	Overseas Containers Ltd.	1973	48, 554	81, 120	8 RND90M	50, 880	MHI Kobe	1981Sep-1982

and with electrical power requirements as low as 300 kW. The Danish industry is also pioneering the reduction of the number of fuel bunker tanks and the use of lagging to decrease bunker oil storage tank heat losses.

For Great Lakes bulk carriers the best solution can sometimes be modification of the turbo-machinery and the replacement of the propeller by a larger unit. In the regard a conversion recently carried out by B&W, Copenhagen is relevant. A twin propeller, 60,000 kW container ship built in 1970 was operating at 40,000 kW by the late 1970's due to the change in world trade. A steam to diesel conversion, costed in Japan, was too expensive and a steam to steam conversion was considered more cost effective. One of the steam turbine sets was removed and the ship converted from twin to single screw. The other steam turbine set was resited on the ship's centerline. The vessel now operates at 20 knot as compared with 27 knots previously. The conversion took two and a half months and the cost was approximately half that estimated for a steam to diesel conversion. The improvement of energy utilization was said to be equal to 15%.

The fuel saving strategy must be designed separately, for the particular vessel, and for a particular trade route.

8.7 Coastal Ships

Profiles of the operation of coastal vessels are generally similar to those of the Great Lakes bulk carriers or ferries. Periods of operation at full speed are relatively short, since distances covered are short and routes are often through confined waters. Therefore, not all fuel efficient measures, which are succesfully employed in ocean going vessels, can give desired benefits when applied to coastal ships. Again, an optimum arrangement must be chosen, bearing in mind all particular conditions of the route and characteristics of the ship.

Some energy saving features of modern coastal ships have been addressed during meetings with European shipbuilders.

In Germany, for instance, all the modern smaller ships, in the 2,000 - 3,500 kW range, burn 180 cSt fuel in the main and auxiliary engines, and shaft generators are often fitted. The only engine burning diesel oil on these ships is the harbour generator set. Waste heat recovery from engine exhaust gases is practiced for heating purposes only. It was stated that the controllable pitch propeller in combination with a medium speed engine is more in demand now for such types of vessel. The small bore long stroke slow speed diesel engines are also very popular. Recently, B & W have sold 43 long stroke 350 mm bore engines, with heavy fuel capability up to 700 cSt, for use in coastal trade vessels.

A good example of a modern, efficient coastal vessel is M.V. ESJA a 1,100 dwt Ro-Ro-Lo-Lo vessel built at Richard Shipbuilders U.K. for Icelandic shipowners [112]. The main features of this ship are:

- Two British Polar medium speed diesel engines capable of burning 180 cSt fuel.

- Power take off from the main engine shaft provides all electrical power at sea.

- Autronica system of control monitoring for unmanned automation.

- Central cooling system with Alpha-Laval plate coolers.

- Anschutz autopilot for accurate navigation.

The most interesting feature of this vessel is a connection in the cooling and fuel heating systems for hot water from geothermal shore supply used while berthed in Icelandic ports. It can be seen from this example, how important is the knowledge and analysis of the ship's operating conditions.

From detailed examination of the intended operating conditions of coastal ships, quite unexpected opportunities and benefits for energy efficiency can be obtained.

8.8 Summary

Of the many energy saving methods open to shipowners some are applicable to all ships while others are only useful for certain vessels. It is most important to detail the operating profile of the vessel before any special device is selected. It is necessary to carry out this exercise before the main propulsion engine and propeller are selected. The usefulness of the various types of energy saving equipment also depends upon whether an existing vessel is to be modified or a newbuilding is contemplated. Table 8.12 lists all fuel efficiency equipment which has been discussed in this report with an indication of the range of fuel savings the equipment may achieve and the suitability of the equipment for different classes of Canadian vessels.

It is not possible to add together all the fuel savings attainable by the various methods described in this study. Some savings reduce the possibilities of further savings while others are necessary to help the attainment of additional improvements. A really fuel efficient ship should be provided with as many combined efficiency measures as possible assuming that thorough economic analysis had been carried beforehand.

Refering to the examples given in Section 8 some observations should be emphasized.

a) The fleet of Canadian lakers offers significant potential for improvement.

b) Steam propulsion, if not changed to diesel should be at least thoroughly investigated in terms of modifying steam plants.

c) Vessels powered by diesel engines should be investigated in order to make conversions enabling burning of heavier grades of fuels. In cases where such conversions are not possible at least fuel blenders may be fitted and used to lower fuel bills.

TABLE 8.12

LIST OF FUEL EFFICIENCY MEASURES APPLICABLE TO CANADIAN SHIPS

NO.	FUEL EFFICIENCY MEASURE	POSSIBLE SAVING IN FUEL COSTS %	POSSIBLE APPLICATION TO CANADIAN SHIPS*			
			LAKER	FERRY	OCEAN GOING	COASTER
	Fuel Quality and Equipment					
1.	Conversion from diesel oil to heavier fuels	<27	N,R	N,R	N,R	N,R
2.	Homogenisation	1	R	R	R	R
3.	Blending	<9	R	R	R	R
4.	Emulsification	<7	N,R	N,R	N,R	N,R
5.	Chemical treatment	<1	N,R	N,R	N,R	N,R
6.	Sludge from centrifuges burned in boilers	0.7	N,R	N,R	N,R	N,R
	Main Diesel Engines					
7.	Use of slow speed, long stroke diesel engine	<16	N	(N)	N	N
8.	Use of medium speed diesel engine	<16	N	N	N	N
9.	Modification of the existing engines to slow steaming	<3.5	R	R	R	R
	Main Steam Turbines					
10.	Re-engining to diesel	0-33	(R)	-	R	-
11.	Retrofitting steam to steam	<5	R	-	R	-
12.	Improvement in steam cycle					
	a) installation of economizer	1-3	R	-	R	-
	b) smaller main feed pump for reduced power operation	0.7-1.8	R	-	R	-
	c) increased number of active feedwater heaters	1.0-3.0	R	-	R	-
	d) cascade turbine bleed system	1.0-5.0	R	-	R	-
13.	Retrofitting of burners to more efficient in existing boilers (main and auxiliary)	<3	R	R	R	R
	Propellers and Stern Form					
14.	Large, slow revving propeller	<12	N,(R)	N,(R)	N,(R)	N,(R)
15.	Controllable pitch propeller	-3-10	N	N	N	N
16.	Reduced propeller roughness	<5.5	N,R	N,R	N,R	N,R
17.	Tip Vortex Free propeller	<24	N,R	N,R	N,R	N,R
18.	Contrarotating propeller	<15	-	(N)	(N)	-
19.	Reaction fins	<8	N,R	N,R	N,R	-
20.	Flow modifying nozzle	8-10	N,R	N,R	N,R	N,R
21.	Ducted propeller	<8	N,R	N,R	N,R	N,R
22.	Semi-ducted stern	<2	N,R	N,R	N,R	N,R
23.	Rudder bulb	<5	N,R	N,R	N,R	N,R

TABLE 8.12 (CONT.)

LIST OF FUEL EFFICIENCY MEASURES APPLICABLE TO CANADIAN SHIPS

			POSSIBLE APPLICATION TO CANADIAN SHIPS*			
NO.	FUEL EFFICIENCY MEASURE	POSSIBLE SAVING IN FUEL COSTS %	LAKER	FERRY	OCEAN GOING	COASTER
	Auxiliary Systems					
24.	Main engine driven generators	2	N	N	N	N
25.	Exhaust gas heat recovery	7.0	N	N	N	N
26.	Jacket cooling water heat recovery	1	N	N	N	N
27.	Charge air heat recovery	1.5	N	N	N	N
28.	Main engine driven pumps and compressors	1	N	N	N	N
29.	Variable speed pumps	1	N,R	N,R	N,R	N,R
30.	Use of thermal fluids for heating pruposes	<1	N	N	N	N
31.	Natural ventilation for the engine room	<1	N	N	N	N
32.	Reducing of bunker tank numbers	0-2	N	N	N	N
33.	Lagging of bunker tanks	0-2	N	N	N	N
	Hull Form					
34.	Model testing in order to improve hull performance	1-8	N	N	N	N
35.	Use of self-polishing anti-fouling coatings	2-7	N,R	N,R	N,R	N,R
36.	Use of high tensile steel reducing weight of ship	<2	N	N	N	N
37.	Use of light alloys	<15	(N)	N	-	-
	Ship Operation					
38.	Speed reduction up to 20%	<30	R	R	R	R
39.	Use of high quality autopilot	0-2	N,R	N,R	N,R	N,R
40.	Weather routing	-3-3	N,R	N,R	N,R	N,R
41.	Draft and trim monitoring	2-4	N,R	N,R	N,R	N,R
42.	Main engine monitoring (also boilers)	2-10	N,R	N,R	N,R	N,R
43.	Fuel monitoring	0-2	N,R	N,R	N,R	N,R

Key to Symbols
* - In newbuilding - N
- For retrofittings - R

( ) - Less likely application
- More likely application

d) There is an overall trend for lake ships to be propelled by slow speed, heavy fuel burning diesel engines rather than by traditional steam turbines. While the slow speed diesel engine provides the lowest specific fuel consumption for propulsion purposes of open water ships it is probable that the medium speed engine in the father-son configuration, will provide the most economical propulsion machinery system for lakers.

e) The principles of using propellers with large diameter, low r.p.m., lower blade number, smaller blade area ratio, smooth blade surface are well established and are being increasingly followed on a world wide basis.

f) In cases where these principles have been applied, such as the Port Weller built CANADIAN ENTERPRISE, the use of a larger propeller used in conjuction with a flow modifying semi-tunnel stern demonstrated fuel saving of 10% over a sister ship with a smaller and higher revving propeller without the semi-tunnel stern.

g) The ducted propeller is a feature finding increasing application in Canadian lakers. Besides having the general benefits as applied on all cargo ships, the ducts used on lakers have an additional benefit. In some applications, steering nozzles have been used thereby eliminating the rudder. This not only improves the manoeuvrability of the ship, it also reduces the length of the ship aft of the engine room, thereby increasing the length of usable cargo space. This is a significant benefit in the case of lakers when maximum lengths are limited by the seaway lock dimensions. For ships with less stringent length limit, the higher capital and maintenance cost of a steering nozzle may make the use of a steering nozzle less attractive.

h) For ship resistance reduction, bulbous bows are increasingly being used and the recently built Great Lakes ocean going bulk carriers are fitted with this bow form. In addition to reducing the hydrodynamic resistance of the hull, improved ice breaking capability is claimed. Obviously, higher ice breaking capability will enable the ship to operate for a longer season, achieve a higher annual return on capital employed and hence be more "efficient".

i) Another method of reducing ship resistance is the use of self-polishing coatings. Due to the presence of ice the application of self-polishing coatings on Canadian ships will differ slightly from ships engaged in open sea trades. As self-polishing coatings have low abrasive resistance, these coatings will perform poorly and give very little protection to the hull steel in the ice belt. To overcome this, a combination of ice resistance coatings (such as Inerta 160 or Zebron) in the ice contacting area and the usual self-polishing copolymer in non ice contacting area will form a good combination.

9.0 FUTURE FUELS AND POWER SOURCES

9.1 General

It has been predicted that natural oil products will be in sufficient supply to meet demand until the beginning of the 21st century, i.e. for 15 years. The balance between the demand and supply of energy will possibly become upset as natural oil becomes more scarce and cause serious disruption of the world's economy if no appropriate remedial measures are taken. The measures which must be taken are to phase in the use of alternative fuels and alternative or more developed technology to permit the traditional oil fuel sources to be phased out.

When considering the future fuel requirements for the ship propulsion, the characteristic needs of ships must be taken into account as follows:

- small power requirements (10 to 30 MW) compared with power plants ashore
- propulsion plants should be relatively compact and light
- propulsions plants should be self-contained
- fuel must be easy and safe to handle, both between shore and ship and between storage and service systems aboard ship
- fuel cannot cause damage to the environment

Based upon information sourced primarily from the technical press, with some recourse to manufacturers for specific data, an evaluation of the leading future fuels for various ship applications has been developed. The intent of the evaluation is to select fuels which can be applicable in future merchant ships, provided the appropriate development work is carried out.

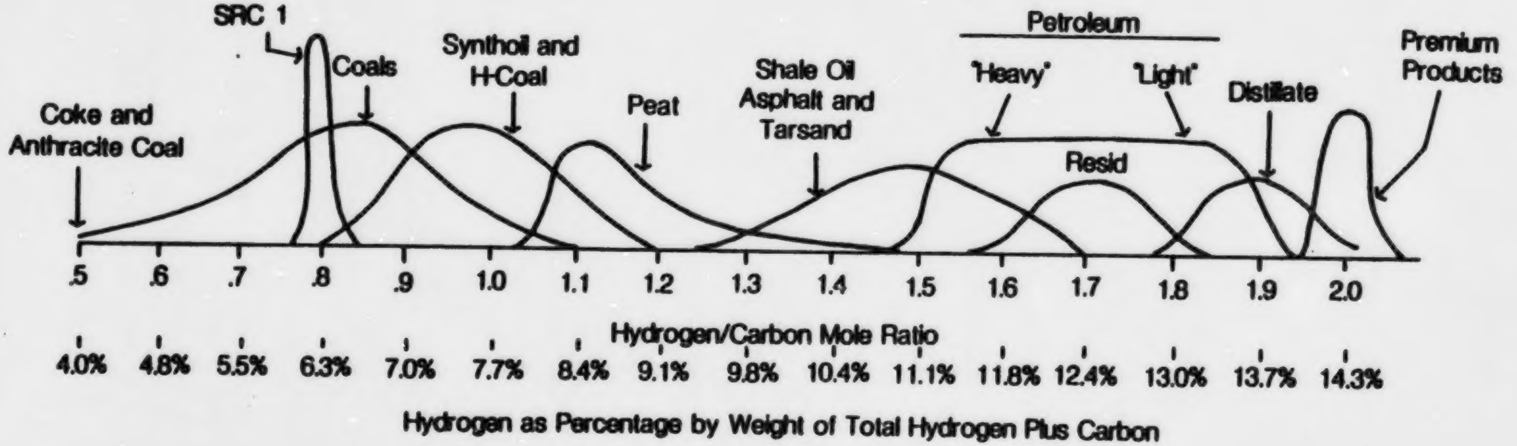
9.2 Synthetic Liquid Fuels

Oil shales, tar sands and coal, are low grade hydrocarbon fuels from which synthetic fuels may be manufactured. The estimated reserves of hydrocarbons in oil shales and tar sands are twice that estimated for the ultimate world recoverable reserves of crude oil. These potential sources of fuel occur in large quantities in Canada, Venezuela, Columbia and U.S.A. The deposits occur as bitumen in a matrix of clay, water and sand and are nearer the earth's surface than petroleum. It is difficult, expensive and inefficient to recover the bitumen from the deposits, although research is being carried out to improve the recovery process. The pace at which research is being conducted has eased considerably in recent years due to the temporary surplus of natural oil.

One of the most important variables which determine the suitability of the combustion properties of tar sands, oil shale and coal is the amount of hydrogen chemically combined with the carbon. Figure 9.1 gives the hydrogen-to-carbon ratio, as percentage of hydrogen by weight, in important carbon based fuels and fuel sources [113].

FIGURE 9.1

HYDROGEN CONTENT OF VARIOUS HYDROCARBON SOURCES AND THEIR END PRODUCTS



[113]

The production of liquid fuel from coal has been carried out commercially for many years. The essential feature of this process is the chemical addition of extra hydrogen to the fuel. During the second World War, the Lurgi process was extensively used in Germany. More recently, the Lurgi process has been further developed in South Africa (SASOL) where up to 38,000 tonne/day of liquid hydrocarbons are produced from low grade coal. Many other methods for providing synthetic liquid fuel from coal have been proposed and pilot plants built, see Table 9.1 [114]. Technology is available to produce most of the range of petroleum products from coal if required although the SASOL/LURGI process is the only full scale plant at present.

One coal conversion process using a minimum of hydrogen is now in development and is known as the short-contact-time SRC (solvent refined coal). The product contains much less hydrogen than current bunker fuels, but does offer a fuel suitable for use by ships fitted with appropriate power plants. These fuels are solids at room temperature, contain more ash than petroleum fuels and can present severe compatibility problems when mixed with petroleum fuels.

The synthetic fuels with higher hydrogen contents are produced in the indirect conversion process where coal is at first converted into carbon monoxide and hydrogen and then the desire products are formed by reaction of these gases. Such products are more applicable to gas turbines and diesel engines.

Oil which is derived from oil shales contain much more hydrogen than coal but less than most crude oil. The best oil shale deposits in North America occur in a small region in the West, where water is limited. As large volumes of water are required to extract and convert the oil from shale, production may be limited. Oil shales can yield excellent marine fuels, but due to their high quality, there will be a strong tendency to upgrade them into such premium transportation fuels as light diesel, gasolines, etc.

The bitumen extracted from tar sands is coked for carbon rejection before further processing and has a reasonably high hydrogen content. After coking, the extracted oils are similar to petroleum crude oil. Canada has large deposits of tar sands in Alberta.

All of these sources of carbon based fuels are contaminated by sulphur, nitrogen and oxygen as well as by metals such as nickel, vanadium and arsenic. These substances usually are present as organic compounds and due to their detrimental effects on the metals in power plants, most synthetic fuels have to be additionally processed. This processing increases the cost.

9.2.1 Synthetic Fuels For Diesel Engines

The essential parameter which permits the combustion of synthetic fuels in a diesel engine, is the cetane number. It is well known that medium and high speed diesels need fuels with a higher cetane number than slow speed engines. Since the cetane number depends mainly on hydrogen content in the fuel, as shown in Figure 9.2, fuels with lower hydrogen content can be used

TABLE 9.1

WORLD DEVELOPMENTS IN LIQUID FUEL FROM COAL PROCESSES

GASIFICATION - SYNTHESIS

COUNTRY	PROCESS IDENTIFICATION AND TYPE	PROCESS	PRODUCTS	STATUS
United States	Gas Synthesis Mobil Oil	Coal Synthesis Gas Methanol - Gasoline	High Octane Gasoline	Laboratory Scale Plant Only.
West Germany	Fischer Tropsch Ruhrkohle A.G.	Lurgi Gasification and Synthesis	Range of Liquid Distillates	New Design Studies.
South Africa	Sasol Sasolburgh	Lurgi Gasification followed by Fischer Tropsch, Arge and Kellogg Synthesis	Gasoline and Chemical Feedstocks	Sasol I Processes about 10,000 Tons of Coal Per Day. Sasol II will process 38,000 t/d.
	AE & CI Modderfontein	Koppers-Totzek Gasification Methanol and Ammonia Synthesis	41,000 t/d Ammonia 60 t/d Methanol	Now Being Built (1980).

TABLE 9.1 (Contd.)

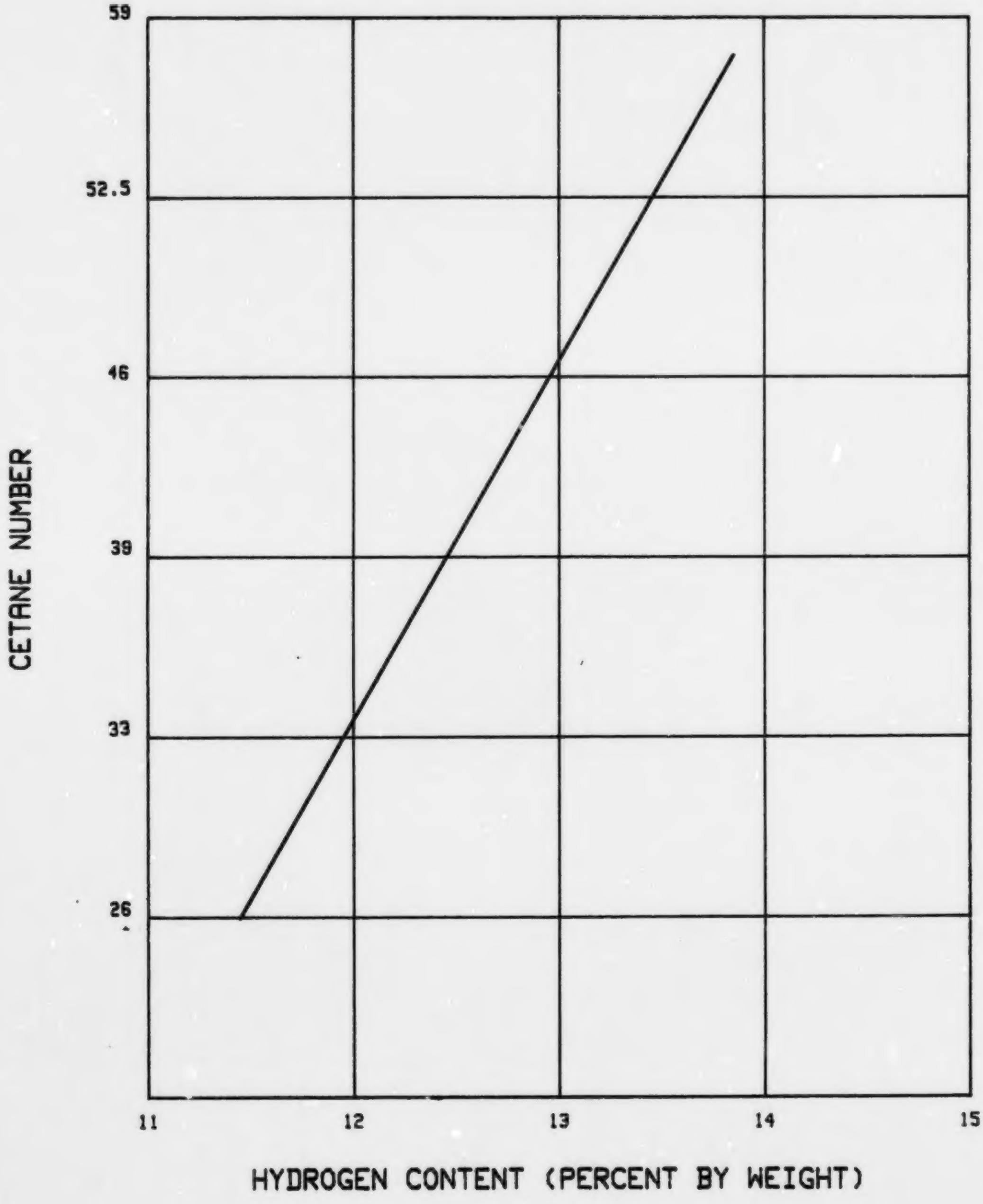
WORLD DEVELOPMENTS IN LIQUID FUEL FROM COAL PROCESSES

LIQUID SOLVENT EXTRACTION - HYDROGENATION

COUNTRY	PROCESS IDENTIFICATION AND TYPE	PROCESS	PRODUCTS	STATUS
West Germany	Bergbau-Forschung (BBF)	Hydrogenation of Coal Slurries Similar to the Bergius Process	Fuel Oils	30 kg/h Pilot Plant Now Operating.
	Saarbergwerke	Similar to BBF Process but with Selective Oil Recycle	Fuel Oils	10 kg/h Plant Now Operating. 6 Plants Planned.
United States	H-Coal	High Pressure (210 Atmospheres) Catalytic Hydrogenation of Coal Slurry in Ebullient Bed.	Refinery Feed 4 bbl/ton	600 t/d Plant Now Being Built (1980).
	Exxon	Coal Hydrogenated During Extraction with Hydrogen Donor Solvent.	Low Boiling Distillate 22 bbl/ton, plus fuel oil.	250 t/d Plant Now Being Built (1980).
	Consol	Coal Extracted with Hydrogen Donor Solvent. Extracted Hydrocracked.	Fuel Oil	20 t/d Plant Operating at Cresap, W. Virginia
	Pamco	Coal Extracted with Solvent in Presence of Hydrogen. Solvent Recycled.	Solvent Refined Coal (SRC) Boiler Fuel Oil	50 t/d Plant at Tacoma, Washington
	Southern Services	Similar to Pamco	SRC Boiler Fuel Oil	6 t/d Plant at Wilsonville.
United Kingdom	Liquid Solvent Extraction (NCB)	Catalytic Hydrocracking of Liquid Solvent extracts from Coal.	Low Boiling Distillate for Automotive Fuels and Chemicals. Total Distillate 60% coal.	30 kg/h Pilot Extraction Plant 2 kg/h Hydrocracking Plant. Designing 1 tph Integrated Plant.
	Supercritical Gas Extraction (NCB)	Catalytic-Hydrocracking of Supercritical Gas Extracts	Similar to Above. Total Distillates 40% Coal	10 kg/h Pilot Plant. Designing 1 tph Integrated Plant.
Poland	Central Mining Institute, Warsaw	Catalytic Hydrogenation of Solvent Extracts from Coal.	Refinery Feed	1 t/d Pilot Plant Operating.
Japan	Cherry Process	High Pressures (High Temp.) Cracking Oil/Coal Mixtures.	Coke and Distillates	24 t/d Pilot Plant Operating.
	Solvent Refined Coal (SRC) Plants Mitsui Sumitomo Coal Taikeryo Coal	Solvent Extraction with Hydrogentation	SRC Fuel Oil	
Russia	Direct Coal Hyrdogenation	Similar to Bergius Process but Using Petroleum Oils as Hydrogen Donor Solvents.	Range of Oils	Small Scale Pilot Plant with Plans for Plant Processing 12 mty

FIGURE 9.2

CETANE NUMBER VS. HYDROGEN CONTENT
FOR A BROAD SELECTION OF DIESEL FUELS



[113]

in slow speed diesel engines while medium and high speed diesels require fuel more enriched in hydrogen. This means that distillates derived from coal are the most difficult to use efficiently in medium and high speed diesel engines, followed by shale products. Fuel derived from tar sands are the most likely to be used in a diesel engine without large modifications being required.

The following problems are relevant to the application of synthetic fuels to diesel engines:

- Engine maintenance may be increased due to the higher ash content, especially with fuel derived from coal.
- Without hydrogen enrichment, synthetic fuels would cause reduction in engine power output.
- Smoke, NOx and particulate output may be increased above safe levels.
- Low cetane number of the fuel will cause increased ignition lag, may cause poor startability, engine roughness and larger amounts of deposits, hence increased maintenance requirements.
- Higher viscosity of synthetic fuels will require modifications to the fuel handling and transport system to be made.
- Additional precautions may be required in the storage of synthetic fuels due to their toxicity.

All of these disadvantages can be overcome by better hydrogenation. Since the addition of hydrogen to the fuel is an expensive process, the quantity of hydrogen added must be regulated by the benefits to be gained.

The latest experiments carried out on slow speed diesel engines have indicated that in spite of the above problems, the slow speed diesel engine could use synthetic fuels from coal when they become commercially available and competitive.

9.2.2 Synthetic Fuels For Steam Propulsion

Any kind of synthetic fuels can be burned in boiler/steam turbine systems provided the problems of environmental contamination by the exhaust gases and onboard fuel handling and transportation can be resolved.

9.3 Coal Fuels

Coal is the most plentiful source of fossil fuel derived energy. The world's total coal reserves are estimated at 4,000 billion cubic meters of oil equivalent (by U.S. Geological Survey). Of these known reserves 48% are in the U.S., nearly 20% in the U.S.S.R. and about 8% in Europe. Although coal reserves are greater than those of both gas and oil and could meet present rates of consumption of all fuels for several hundred years, its use has decreased. The decrease in the consumption of coal is for many reasons, not least among which are; oil is more versatile; is less of a pollutant when burnt; is more easily stored and transported; and requires less capital intensive equipment to release the energy.

Coal can be used directly as a ship's fuel in stoker-grade grains or in a pulverised form as well as in mixtures with oil, as the so called coal-oil slurries. Although there is a penalty in either cargo capacity or ship size while using raw coal as fuel, the fuel cost savings far overshadow the loss of cargo on routes where coal can be purchased cheaply. Many ports of the world have facilities for handling cargo coal in bulk, however few ports have any facility for handling stoker-grade bunker coal and this is one restraint on the present widespread use of coal as a marine fuel.

Table 9.2 shows typical properties of a range of coals. It will be noticed that specific gravity, fixed carbon content and calorific value increase with rank, and that water content and volatile matter decrease with rank.

Bituminous coals are the most abundant and, together with sub-bituminous coals, are those usually favoured for use in marine boilers. Lignites are used sometimes but the ash yield is high and spontaneous combustion is more likely.

In order to overcome difficulties with coal handling in the solid form, efforts have been made to develop means to make coal behave like a fluid. This has as been achieved by pulverizing the coal to turn it into the colloidal or slurry form. Usually coal/oil mixture consists of 30 to 50% by weight of pulverized black coal suspended in heavy petroleum oil, with or without stabilising additives. The idea is to maintain many of the handling, storage and combustion properties of oil while taking advantage of the lower cost of coal. The typical coal particle sizes are 50 to 75 microns but much smaller sizes down to 4 microns are also used. A concern is to ensure that the coal/oil mixture maintains homogenity and the coal particles do not settle out under the influence of gravity.

Coal/oil mixtures have a viscosity higher than the oil and the viscosity varies greatly according to the coal grain size and the coal/oil mixture ratio as shown in Figure 9.3 [114]. Viscosity of slurries increase dramatically when the coal content exceeds about 45%.

Reduction of coal particle size also improves combustion characteristics but at the cost of increased energy for grinding. The reason why the optimum coal/oil ratio is approximately 40% is explained in Figure 9.4. Under the fixed conditions of coal, oil and temperature, good long term stability can be exhibited only in mixtures approaching 40%. Some processes which are used to make coal/oil slurries, grind coal after mixing with the oil. This is reported to produce much more stable mixtures. Another process makes a slurry by mixing oil, coal and water by ultrasonic agitation, where the water mixed with the oil acts as a stabilizer.

As particles tend to agglomerate, thus accelerating the settling process, stability of the slurry is adversely affected. It is believed that constant agitation, perhaps by a forced recirculation system will be necessary to maintain the coal in suspension.

TABLE 9.2

TYPICAL PROPERTIES FOR A RANGE OF COALS

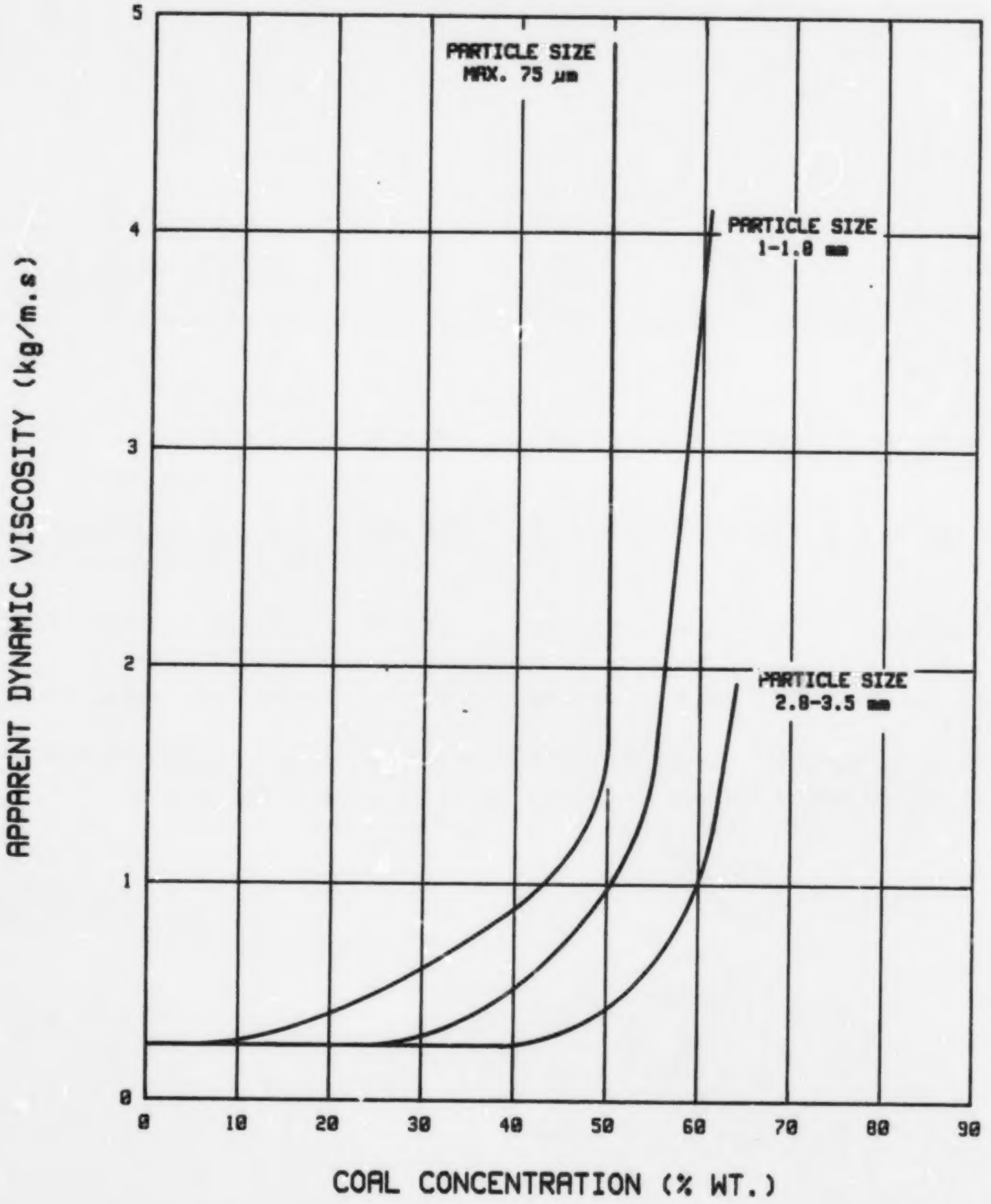
RANK	SPECIFIC GRAVITY	WATER CONTENT %	FIXED CARBON %	VOLATILE MATTER %	H. CALORIFIC* VALUE (kJ/kg)
Anthracite	1.4 - 1.8	about 3	92 - 98	2 - 8	29300 - 33500
Semi-Anthracite	-	3 - 5	86 - 92	8 - 14	35000
Bituminous	1.2 - 1.5	3 - 15	46 - 86	25 - 40	27200 - 31400
Sub-Bituminous	1.2 - 1.4	15 - 25	37 - 45	33 - 50	14700 - 27200
Lignite (or Brown)	1.1 - 1.3	20 - 50	27 - 31	25 - 55	8400 - 16800

* By comparison the standard higher calorific value of fuel oil is 44,965 kJ/kg. (Equivalent to LCV = 10,200 kcal/kg).

[115]

FIGURE 9.3

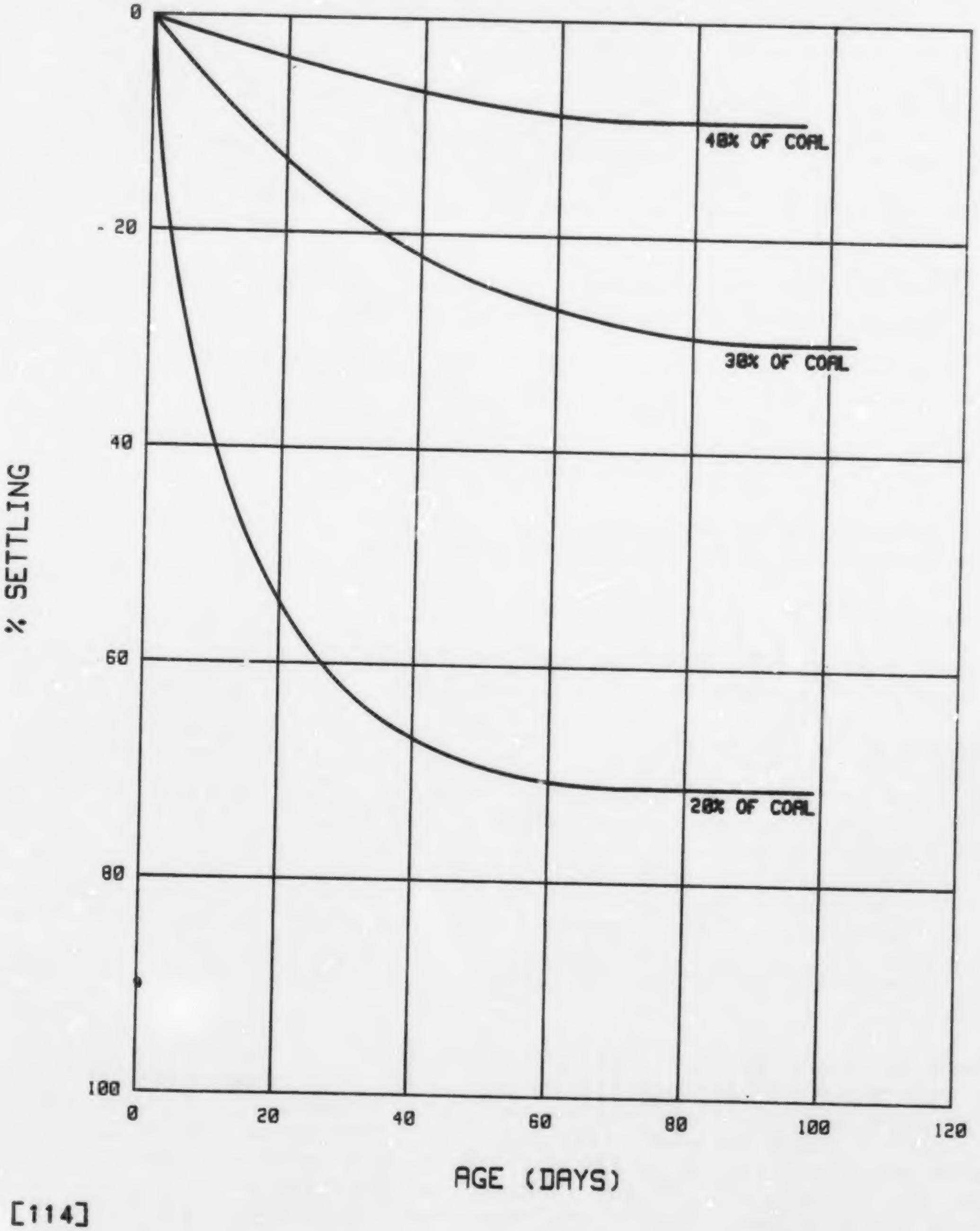
EFFECT OF COAL CONCENTRATION AND PARTICLE SIZE ON VISCOSITY OF COAL/OIL MIXTURE



BITUMINOUS COAL No 6
FUEL OIL TEMP. 25°C [114]

FIGURE 9.4

EFFECT OF PERCENTAGE OF COAL ON STABILITY OF COAL/OIL MIXTURE



[114]

COAL SIZE - MAX 66μm OIL - No 6 FUEL OIL
TEMPERATURE - 25°C

There is also a problem of quality of coal/oil mixture. Although no representative data is available of density, viscosity, ash content, pour point, etc. for coal/oil slurries, it may be inferred that the burning characteristics, carbon residue, ash content and pour point of slurries will be worse than those of the unadulterated fuel oil. As regard to fuel treatment, it will not be possible to remove fine particles from slurries by centrifuging, so that it will be necessary to perform this operation on the fuel oil before mixing with the coal.

Coal/oil mixtures would have substantial advantages over coal if the price of fuel oil, after allowing for the difference in calorific value, rises well above that of coal. Some of these advantages would be:

- bunkering on board ship will be easier with slurries than with coal
- handling problems associated with coal such as storage and ash disposal would be largely avoided
- existing liquid fuel handling and storage arrangement may be usable with minor modifications
- the diesel engine is likely to be more readily adaptable to burning slurry than solid coal

All these potential advantages should be taken into account while planning the areas for research of alternative fuel to oil.

9.3.1 Coal Fuels For Diesel Engines

The utilization of pulverized coal and coal/oil mixtures in dierel engines is confronted by some technical difficulties.

Burning of pulverized coal in the cylinders of an engine would require conversion of the liquid fuel injection system and the fitting of a dust-aspiration system. The dimensions of the exhaust valves or ports would need to be enlarged as more fuel must be injected into the engine to achieve the required horsepower. Apart from problems in storage and handling, this fuel would most probably cause excessive wear and erosion of cylinder liners, exhaust valves and ports as well as fouling of turbo-chargers and heat recovery equipment. These are only some of the obvious problems to be overcome and will test the ingenuity of the engine designers and make the development of a practical solid fuel diesel engine unlikely in the near term.

Much better immediate prospects may be expected from the development of diesel engines to burn coal slurries. Continuing research by the leading designers of large diesel engines confirms confidence in the diesel engine's ability to burn the coal based slurries. Tests conducted by Sulzer with slurries containing up to 50% coal and a newly developed injection system for the admission of such fuels into the cylinder has proved the slurries to be a suitable fuel. Due to the extremely high ash content (silicate ash content by weight nearly 10%), extreme rates of erosion of the injection nozzles, as well as wear of the piston rings and liners were experienced. Consequently, it would appear necessary to reduce the ash content to more reasonable limits prior to combustion.

The process of ash separation could be effected during the final grinding of the coal dust. Such coal beneficiation processes are already being operated successfully in pilot plants on shore. The wear resulting from the residual contaminants must be reduced through technical improvement of the engine components. Tests have shown that the best substitute for the heavy oil burned today would be micro-slurry (5 micron coal particles) with a composition of 50% coal, 20% oil and 30% water, by weight [116].

9.3.2 Coal Fuels For Steam Propulsion

Coal fired steam boilers are the only readily available technology for directly utilizing coal for power generation. All forms of coal can be burned in boilers, although for the best efficiency boilers must be designed for specific types of coal. The boilers may be fired by a variety of methods, namely:

- coal slurry firing
- pulverized coal firing
- solid coal firing by means of mechanical stokers or fluidized bed combustors.

Coal slurry firing of boilers has the advantage that the handling and burning equipment will be similar in most respects to oil firing. However, the processing costs for the coal slurry and lack of bunkering facilities makes this alternative less attractive, at least at present. The method may hold some potential for converting oil fired boilers to coal slurry burning although this type of conversion may not be economically viable for many years.

Pulverized coal fired boilers are a real alternative for marine use. However, the space requirements, higher capital costs, the limitation on the range of coals which can be used, together with the potential safety problems related to operation of the pulverizer and storage of pulverized coal, makes this type of firing a less attractive alternative than stoker fired boilers. Stoker firing is the most suitable for small boilers and permits a large variety of coals to be burned efficiently with a minimum of sophisticated equipment. This type of boiler together with fluidized bed boilers will be discussed in greater detail in Section 10.0.

9.4 Alcohol Fuels

Two types of alcohol have been considered as possible future fuels; methanol or methyl alcohol (CH_3OH), and ethanol or ethyl alcohol (C_2H_5OH). Both alcohols originally were produced from bio-mass; methanol by destructive distillation of high cellulose products such as wood, and ethanol by fermentation of starch or sugar based organic substances. They can also be made almost entirely synthetically.

Methanol is now produced either from naphtha, natural gas or coal and is presently used as chemical fuelstock and not fuel. Methanol can be transported by ordinary tankers. However, methanol plants are costly to build and operate and, due to lack of demand for methanol as a fuel, no plants are expected to be in service in the near future. Besides the high cost of production, methanol has relatively low heating value compared to petroleum derived liquid fuels, and is also more explosive, thus storage and handling systems must be specially designed [117].

Various exaggerated estimates have been made of the production potential of ethanol over the past ten years, but it is now generally accepted that the resource base is limited. The economic justification for the Brazilian attempts at ethanol production in recent year by the use of bio-mass was based on an abundance of cheap labour. Those experiments have been fraught with technological problems and the economics have been less favourable than originally contemplated. Ethanol can also be produced synthetically from ethylene or from coal. Its properties are similar to those of methanol [118].

9.4.1 Alcohol Fuels For Diesel Engines

The low cetane number of both ethyl and methyl alcohols make their use in diesel engines difficult. However, poor ignition quality can be counteracted through the following methods:

- Burning the alcohol as a mixture with another fuel of appropriate cetane number. The change of cetane number according to ethanol content in diesel oil is shown in Figure 9.5 [113].

- Introducing the alcohol separately by dual injection or carburetion to be mixed and burned with a pilot charge.

- Burning the alcohol as a homogeneous emulsion with another fuel of high ignition quality, the emulsion to be produced in a suitable device just prior to injection.

Utilization of these methods is limited by the following problems:

- Very poor compatibility of alcohols with petroleum fuels, especially at low temperatures or in the presence of water. Methanol for instance is not miscible in diesel fuel at average ambient temperature as shown in Figure 9.6 [113].

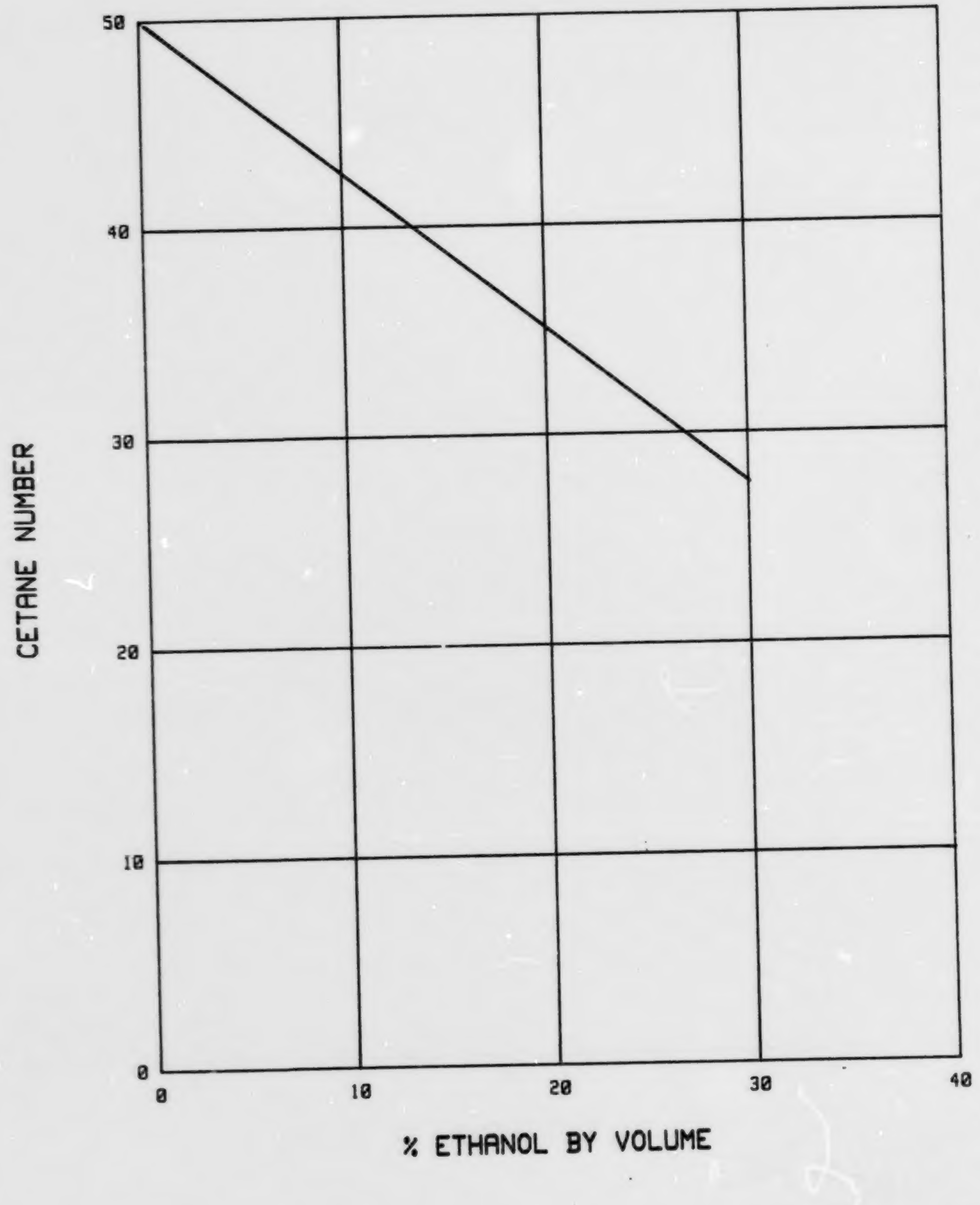
- Alcohol-diesel oil blends display gum forming tendencies even when using stabilisers.

- The use of two fuels implies difficulties in storage and handling. This is magnified by the need of increased bunkering frequency or greater storage capacity due to the lower heat content of alcohols.

Both slow speed and medium speed diesel engines, would require considerable redesign to permit alcohol fuels to be used. Modifications would be needed

FIGURE 9.5

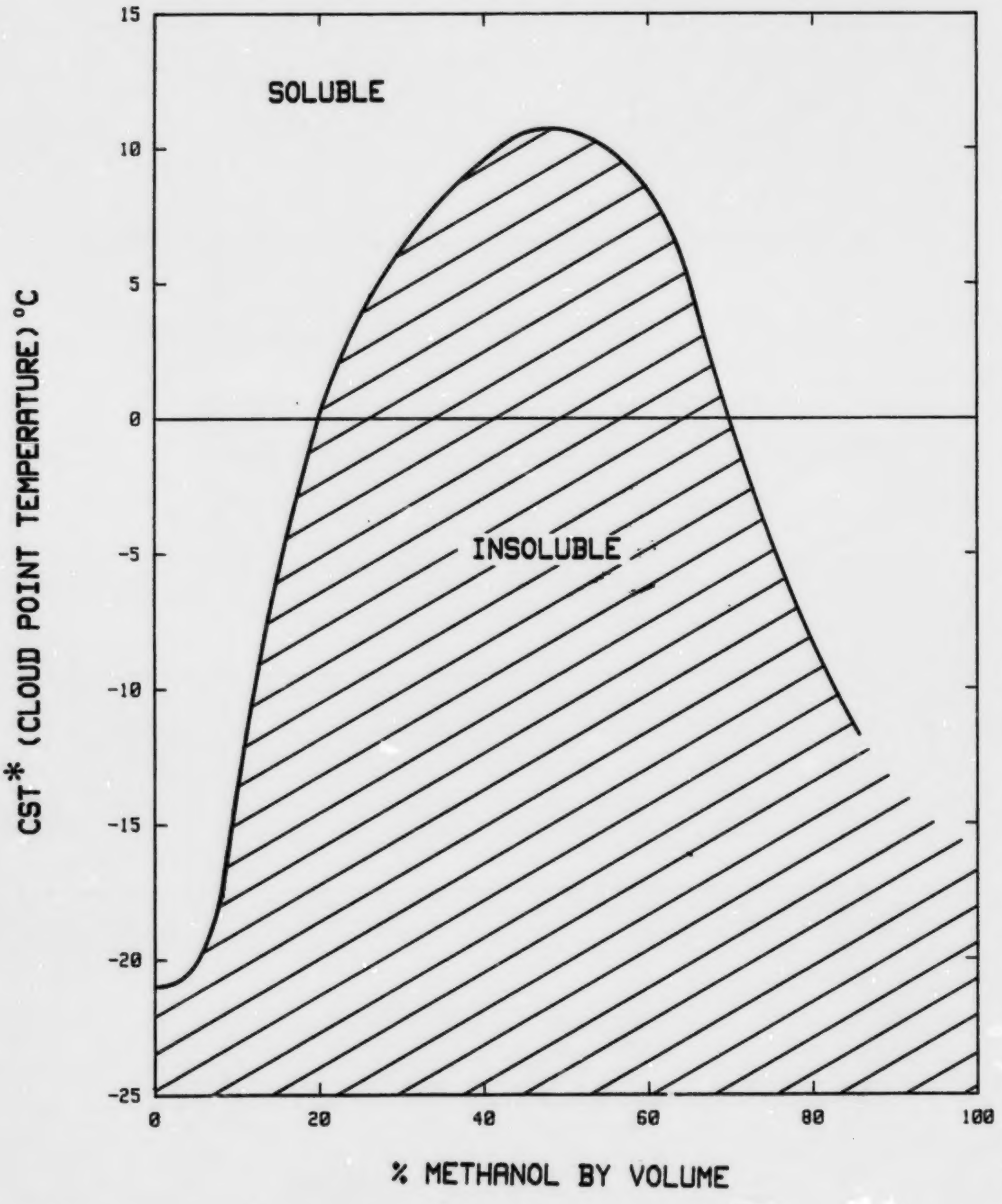
EFFECT OF ETHANOL CONTENT ON CETANE NUMBER OF DIESEL OIL / ETHANOL MIXTURE



[113]

FIGURE 9.6

SOLUBILITY OF METHANOL IN A No.2 DIESEL FUEL



[113]

*CRITICAL SOLUTION TEMPERATURE

to the combustion chamber and fuel injection system as well as to fuel handling and supply systems due to explosion hazards.

In the case of trunk piston engines, increased wear and corrosion of the engine due to the dilution of the lubrication oil by the alcohol may cause problems. These difficulties are not insurmountable but require development of engine systems specific to those fuels.

In view of the disadvantages, all of which can be overcome at a cost, it is unlikely that these fuels will be adapted to diesel engines in the near term.

9.4.2 Alcohol Fuels For Steam Propulsion

Alcohols can easily be burned in steam boilers but due to the high production cost of the fuel it is unlikely that they will be used in place of the lower grade fuels. However, alcohols may be suitable for use in boilers in the form of an alcohol/coal slurry. The technology required to produce methanol/coal slurry is now available, but has not been applied to large scale production.

9.4.3 Alcohol Fuels For Gas Turbines

Alcohols and their blends can be burned in gas turbines without difficulty since gas turbines are insensitive to cetane number. Some problems can occur due to the alcohol's property to pick up and retain water. In the case of salt water content in the alcohol, this can lead to corrosion in the hot sections of the turbine. This would only be a problem if oil/alcohol blends were burned, since alcohols are in themselves pure and do not contain harmful constituents such as sulphur and vanadium.

9.5 Gaseous Fuels

Gases of various kinds have been used as an engine fuel ashore and some of these gases may also be considered as the potential fuel for marine propulsion engines. The most important fuel gases which are already used or can be used for engine fuel are gaseous hydrocarbons, such as methane, propane and butane, natural gas and hydrogen.

- Methane (CH_4) is the main component of natural gas (in some cases up to 98.5%). The other production methods of methane include the anaerobic decomposition of vegetation, gasification of coal and synthesis of carbon monoxide and hydrogen.

- Propane (C_3H_8) and butane (C_4H_{10}) are extracted from natural gas or from petroleum, in which they are dissolved, and stored under pressure. In this form they are known as liquefied petroleum gas (LPG) or bottle gas.

  This mixture has found a successful application for mobile plant in factories, for spark ignition engines of motor cars and other places where a clean exhaust is essential. At present these "heavy" natural gases are not permitted by regulation to be used as fuel gas on board ship. These gases are heavier than air and if a leak develops, gas pockets can accumulate with the danger of explosion.

- Natural gas has considerable use in shore based industries. The use of natural gas at sea is confined to LNG (liquefied natural gas) vessels with the exception of one Australian coastal vessel which utilizes compressed natural gas (CNG) as the main fuel in a dual fuel internal combustion engine. The composition of natural gas depends on its source. The average composition by volume is: methane 92%, ethane 3%, propane 2% and nitrogen 3% [119].

  The world's reserves of natural gases are estimated to be more than 70% of those of oil although their development and utilization have been generally less advanced than oil, except in the United States. Maximum gas production is expected to peak 10 to 20 years later than oil. Large quantities of natural gas are found in Canada and its low price makes natural gas one of the most promising alternate energy resources.

- Hydrogen is now produced by steam reforming or partial oxidation of hydrocarbons, particularly natural gas. In the longer term, however, large scale production will have to be based on the dissociation of water either by electrolysis or by direct dissociation into hydrogen and oxygen using solar and ocean thermal energy, nuclear and perhaps tidal and wind energy.

  Hydrogen is an ideal fuel in many respects, but there is a problem in storing the gas in the liquefied state. The liquefaction process itself is very expensive, not only in capital cost but also in energy. With current technology the thermodynamic efficiency of the total liquefaction complex is 23.7%. The liquefaction temperature of hydrogen is 20.3° K at standard pressure. The liquefaction of the gas requires 30 to 49% of the hydrogen to be used to provide the necessary energy. This results in a 30 to 40% increase in cost of liquefied hydrogen (LH_2), as compared with hydrogen gas.

  The technology of LH_2 storage has been well developed for the U.S. space program. This knowledge can be applied for the carriage of LH_2 by sea. Containments of the supercold LH_2 can be similar to the existing LNG ship's containments with minor modifications in insulation and materials. Because of the very low ignition energy of hydrogen and a wide range of flamable hydrogen/air mixtures, safety problems will be severe in future ships which utilize hydrogen as fuel [117].

  Another possible use of hydrogen as a fuel can be achieved indirectly by burning hydrogen combined with a carrier such as carbon or nitrogen. Ammonia (NH_3) and hydrazine (N_2H_4) are fuels classified as hydrogen carriers.

9.5.1 Gaseous Fuels For Diesel Engines

The composition of a gas plays a very important role in its performance as an engine fuel. Generally, gas fuel for compression ignition reciprocating engines must have high anti-knocking properties to permit the use of a high compression ratio to attain the best efficiency. Figure 9.7 shows efficiency plotted against compression ratio for theoretical air cycle and also shows the curve based on the brake thermal efficiency observed in actual engines.

The compression ratio of a diesel engine designed to burn gas is lower than the equivalent liquid fueled engine. This is because gas tends to detonate in the cylinder when the compression ratio is high. This detonation will result in damage to the engine if not contained. In the case of a medium speed engine the usual compression ratio when operating with liquid fuel is about 14:1 whereas the ratio is limited to about 11:1 - 12:1 when designed to burn gas. The limiting compression ratio will result in a gas burning engine having a lower efficiency and lower power output than an equivalent diesel fuel engine. It should be noted that due to the low cetane number or diesel index of natural gas, a pilot injection of diesel or suitable liquid fuel in the cylinder is required. This pilot injection usually represents about 5 to 7% of the fuel to be burned.

The factor which was established for comparing anti-knocking propeties of gases, is the critical compression ratio. It is the compression ratio at which the detonation just begins. The values of the critical compression ratio for various gases listed in order of their octane number are as follows:

Methane (low octane number)	12.6
Ethane	12.4
Propane	12.2
Iso-Butane	8.0
N-Butane	5.5
Iso-Octane (100 Octane No.)	7.3
Ethylene	8.5
Hydrogen	8.2

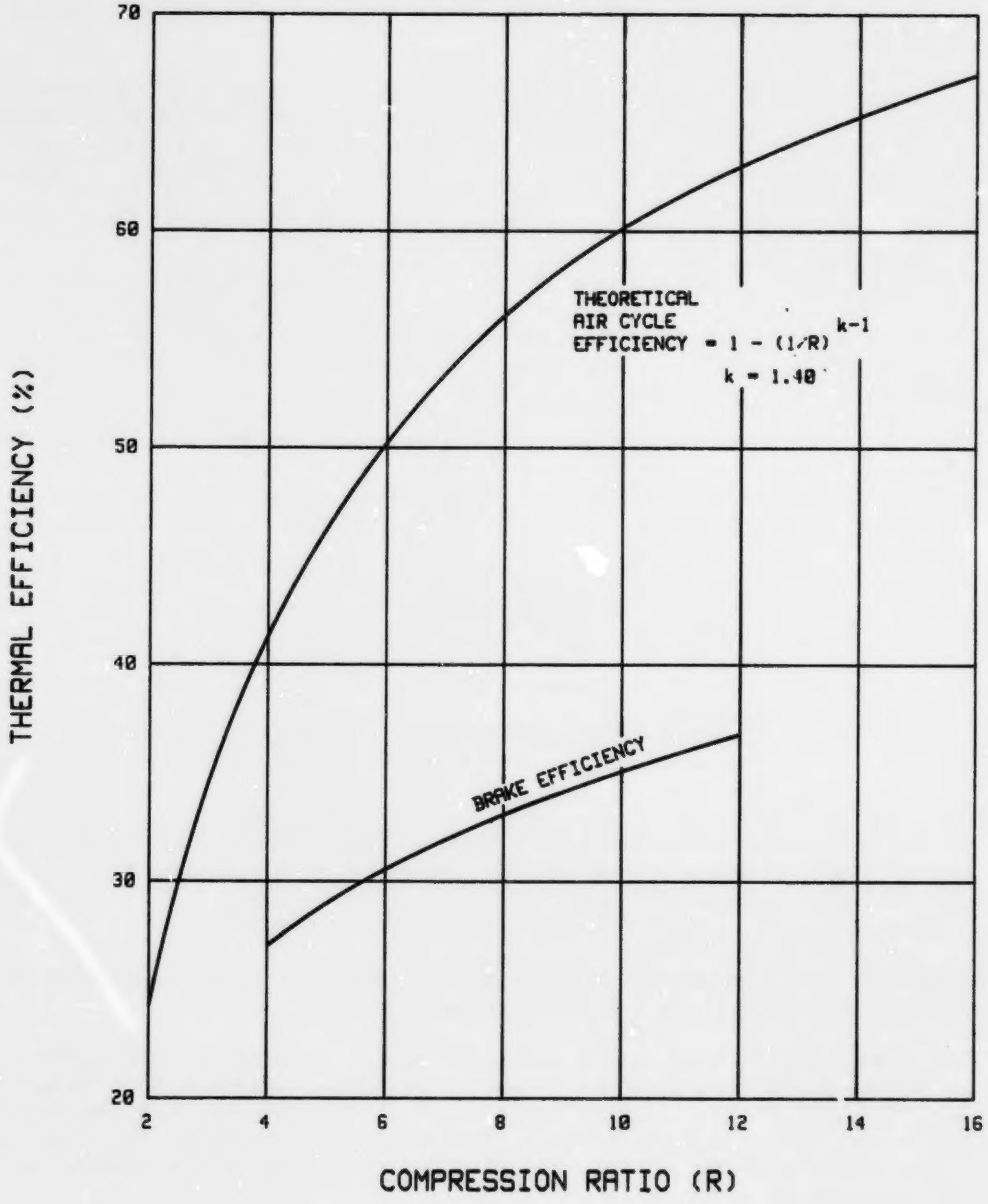
It will be seen that almost all gases listed have an anti-knock rating better than 100 Octane petrol [119]. Note:- high octane fuels are the most suitable for spark ignition engines while low octane fuels (high diesel index) are more suitable for compression ignition machines.

Generally gases which are rich in methane are less prone to detonation and hydrogen rich gases are especially susceptible.

The presence of inert gases such as carbon dioxide or nitrogen in the cylinder charge reduces the flame speed and therefore the tendency to detonate, but it also reduces the calorific value of the fuel and therefore the power output of the engine. The calorific values of the fuel gases differ but the value at the theoretical air fuel ratio are almost the same for each gas as shown in Table 9.3 [120].

FIGURE 9.7

DIESEL CYCLE THEORETICAL EFFICIENCY AND BRAKE EFFICIENCY RELATED TO COMPRESSION RATIO



[119]

TABLE 9.3

CALORIFIC VALUES OF GASES AND THEIR AIR MIXTURES

GAS	NET CALORIFIC VALUE kJ/m^3*	MIXTURE GAS/AIR CALORIFIC VALUE kJ/m^3*
Methane	35,824	3,406
Ethane	64,400	3,645
Propane	93,646	3,775
Butane	121,719	3,808
Natural Gas	35,615	3,419

[120]

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* At normal ambient conditions, i.e. 1.013 bar/0°C

It can be seen from Table 9.3 that all fuel gases which are presently available are approximately equivalent, with respect to the energy available in a combustible mixture in an engine cylinder. From consideration of all facts methane gas and hence natural gas, which is almost pure methane, is best suited to burn in internal combustion reciprocating engines due to its superior anti-knocking properties which permits the engine to achieve the best efficiency.

The design of a gas engine is determined by the self ignition temperature of the gas to be burnt. The self ignition temperature of gaseous fuel is much higher than that of liquid fuels. Therefore either the compression ratio must be increased, ignition achieved by injection of pilot fuel or the use of a spark ignition adapted to ignite the fuel. As an increase of the compression ratio is not possible due to detonation, there are two designs of gas engines:

- dual fuel engines
- spark ignition engines

On the latter, spark ignition system, which is identical to the gasoline engine in its mechanism, the fuel is ignited by electrical discharge in the cylinder by an ignition plug. When the engine is of larger capacity, knocking tends to occur. This system is efficient for lower compression ratio, allows wider selection of gases and can be operated on gas only.

The dual fuel engine is essentially a diesel engine using a pilot injection of diesel fuel to induce ingnition. The engine burns both gas and diesel fuel in varying proportions and will operate wholly on diesel fuel if required. The lower diesel limit for pilot injection is approximately 5 - 10% of diesel fuel depending upon the design and size of the engine. With the present design, an engine operating on diesel fuel may be derated by approximately 10% when operated on natural gas. Not all engines can burn natural gas throughout their operating range and often the ratio of gas to fuel is load dependent, the most gas being consumed at the highest engine load.

The thermal efficiency of dual fuel engines depends greatly on the way in which the gas is admitted to the cylinders. All commercial applications at this time utilize low pressure to admit gas to the cylinders at the start of each compression stroke. The newly developed prototype engines, however, employ the diffuse combustion system. The system injects the highly pressurized gas fuel into the cylinder to obtain the same diffusive combustion process as the conventional diesel engine. With this method, detonation which occurs in conventional gas engines can be avoided in spite of using gas fuel. As the result, higher compression ratio can be used and the thermal efficiency and power rate of this engine are equivalent to that of the conventional diesel engine [121].

The dual fuel engine is suitable for marine duties provided sufficient storage capacity for this gas is available onboard ship. The difficulty of storing gas in sufficient volume and the absence of a suitable supply of gas at ship terminals are the principal reasons why these engines were not previously fitted to ships. The areas of possible application of these engines are limited to:

- LNG carriers
- Ship on dedicated short routes, e.g. ferries

LNG carriers carry gas in the liquid form at -162°C and use the boil-off gas from the cargo to fuel steam boilers with a plant thermal efficiency of not greater than 30%. Application of the gas to dual fuel reciprocating engines for LNG carriers' propulsion system can increase the thermal efficiency to close to 50%.

Ships operating on short routes can utilize compressed natural gas (CNG) stored in special pressure vessels. The technology for operation with CNG is already in use on one ship. The fuel storage capacity onboard limits cargo space and the requirement for dedicated refueling ports limits the range of the ship. It is therefore appropriate that the application be initially investigated for ships on relatively short sea passages. Canadian ferries, coastal vessels and lakers may therefore be prime contenders.

9.5.2 Gaseous Fuels For Steam Propulsion and Gas Turbines

Gaseous fuels can be easily burnt in the furnaces of steam boilers. LNG carriers use natural gas to fuel their boilers and no major problems should be foreseen with the same use of hydrogen. Existing water tube boilers are available with alternative firing equipment for natural gas and liquid fuel, the change over being made without interruption to steam raising. A combination burner is so designed that the flame shape and intensity is similar for either fuel and no difference in boiler performance can be observed.

Gaseous fuels are also ideal fuels for gas turbines of all types. The most likely application of gaseous fuels to gas turbines can be expected on liquefied gas carriers with COGAS machinery.

9.6 Wind Energy

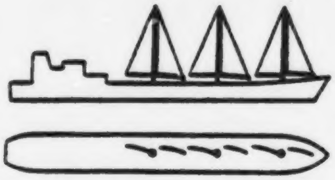
Research into the wind propulsion of ships is under way in several countries. In view of the escalating cost of marine fuel oil, commercial utilization of this "free" source of energy is attractive. The research and development carried out to date is quite varied, and several small ships have been fitted with modern sails for evaluation in service. These experiments have recorded successful results in fuel savings and the experience has been sufficiently encouraging to expect "sail power" to be extended to larger ocean going vessels in the future.

Wind power can be used to assist ship propulsion through the use of sails, wind turbine, Flettner rotors, or even kites. Thrust assistance devices like sails provide thrust without mechanical connection to the main propulsion system. Energy conversion systems, such as wind turbines or windmills, can augment existing power by being "clutched in" to the main propulsion system by electrical, mechanical or hydraulic means.

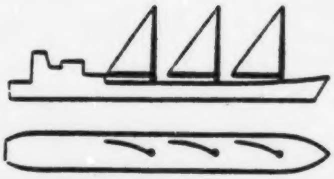
Figure 9.8 shows some types of wind assisted ships. These alternatives have been investigated in the U.S., the results of which have been shown in Figure 9.8.

FIGURE 9.8

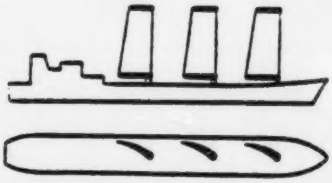
TYPES OF WIND ASSISTED SHIPS AND RELATIVE MERIT RATINGS



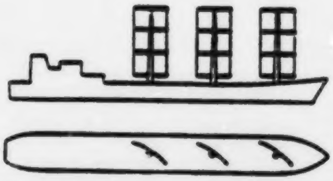
A. STAYED FORE AND AFT RIG



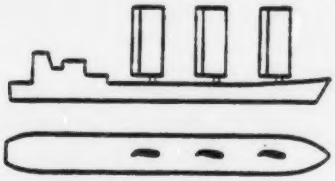
B. UNSTAYED CAT RIG



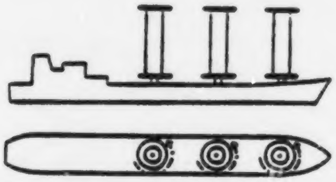
C. PRINCETON SAILWING



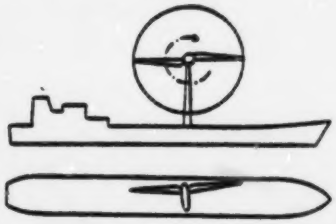
D. SQUARE RIG



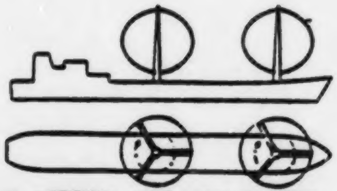
E. WING SAIL



F. FLETTNER ROTOR



G. HORIZONTAL AXIS WIND TURBINE



H. VERTICAL AXIS WIND TURBINE

RIG	PROPULSIVE PERFORMANCE	INITIAL COST	OPERATING COST	WEIGHT	SIZE	RELIABILITY	INTERFERENCE WITH CARGO	OVERALL RANK
Unstayed Cat Rig	Best	Low	Low	Medium	Large	Good	Minimal	1
Wing Sail	Best	Low	Low	Low	Medium	Good	Manageable	2
Square Sail	Good	High	Medium	High	Medium	Poor	Minimal	3
Stayed Fore-and-Aft Rig	Good	Lowest	Low	Medium	Large	Medium	Major	4
Princeton Sail Wing	Good	High	Low	Low	Medium	Good	Minimal	5
Flettner Rotor	Unclear	Low	High	Low	Small	?	Minimal	6
Wind Turbines (Horizontal or Vertical Axis)	Poor	Very High	Very High	High	Very Large	None	Minimal (HA) Major (VA)	7

[28]

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To design wind assisted ships which could be really competitive with conventional vessels the following considerations and requirements must be taken into account:

- Economic viability, i.e., substantial fuel savings should be obtainable.
- Rugged reliability in continuous service at sea.
- Usefulness of the sail assist will depend on trade routes and prevailing wind patterns. The North Atlantic and the North Pacific routes have the highest average wind speeds, offering the best prospects for the highest gains.
- No extra crew would be needed to operate the ship. Through the application of computerized technology and remote control of sails, this requirement can be achieved.
- Rigging must not unduly obstruct cargo handling areas, hence the number of masts should be minimized.
- Assistance devices must be of considerable physical dimensions because of the low grade nature of wind energy.
- Their size implies significant weight which has an influence on stability and cargo deadweight.
- When inoperative under unfavourable weather conditions they will increase air drag and consequently the power required.
- An appropriate deeper ship draft should be provided in order to counter the side thrust of the sails when sailing in the wind.

Apart from these problems, which inhibit the development of wind propulsion, some advantages of modern sails should be noted. Besides the evident fuel saving by augmentation of the thrust for ship propulsion the other advantages of assisting sails are:

- Reduced ship rolling when under sail in a seaway. This reduces the considerable fuel losses incurred through rolling.
- Ships with high superstructures aft, such as modern tankers and bulk carriers, act as if they were carrying a large mizzen sail. They have, therefore, a pronounced tendency to put their head into the wind. In order to keep on a straight course in a beam wind, considerable rudder must be applied. This creates a permanent drag and slows down the ship. A sail area forward would balance this tendency.
- The main engine benefits from an auxiliary sail since it now needs less torque to produce the same speed. This reduces the mean effective pressure and, consequently, wear and maintenance expenses, and of course fuel consumption.

Several studies have been carried out in different countries into the benefits of modern sails. The results from the first successful attempts of utilization of modern sails have revealed that wind assisted propulsion power may be economically feasible for many types of cargo ships. Small and medium sized tankers, as well as fishing vessels were deemed the best possibilities for sail assisted propulsion, but general cargo ships were also found to have economic potential. The savings in total annual shipping economy based upon 1983 fuel prices, taking into account the cost of financing, profits etc. is claimed to be 5-15% for small to medium sized sail assisted merchant vessels and 15-25% for bigger ships of the range 2,000-40,000 dwt [10].

The more traditional sail forms develop the greatest direct thrust on the ship's hull at any wind speed providing the wind angle relative to the bow is greater than seventy degrees, and therefore are often considered as the most advantageous wind assisting devices. Some other studies and trials claim that better economic results can be achieved by means of wing or rigid airfoil sails.

The Flettner rotor using the Magnus effect was successfully demonstrated many years ago. Present day technology now permits this principle to be utilized as a practical means of ship propulsion. The mechanism by which wind blowing onto a driven rotating cylinder is converted into a force capable of propelling a ship is illustrated in Figure 9.9. The drag to lift ratio is not much better than that of conventional sailing vessels, but the "rig" is much more compact and also very much less complicated mechanically.

A wind turbine or windmill can be used to propel a ship in a number of ways. The turbine can extract wind energy and use it to propel the ship via a conventional propeller, or the turbine can be allowed to freely rotate under wind action to generate an aerodynamic lift which propels the vessel. There are also more complicated combinations. Performance estimates for this propulsion system are incomplete and there would be considerable engineering problems in applying such a system to a ship. The system's primary claim for serious attention is that it is omnidirectional and therefore better adapted to navigation in restricted channels by sailing directly head to wind.

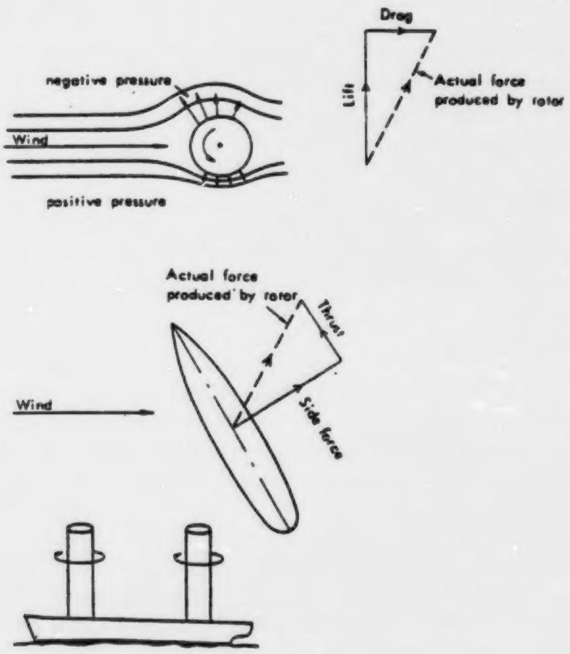
The efficiency of wind assisted propulsion is illustrated by the following examples of existing vessels:

- SHIN AITOKU MARU, a 1,600 dwt coastal tanker is the world's first commercial motor tanker equipped with sails as an auxiliary propulsion device. Two rigid airfoil sails fold around a vertical mast. The mast can be slewed around on turntables to adjust the airfoils to the correct angle of attack. The setting and fold of the sails are by hydraulic actuators and controlled by a computer on the bridge. The ship operates in the coastal waters of Japan.

  According to the results from actual voyages, the time during which the sails were used accounted for an average of 60% of the total voyage hours. The saving of fuel realised by use of the sails, was calculated at 8 to 10%. The main engine output is

FIGURE 9.9

THE FLETTNER ROTOR -
PRINCIPLE OF OPERATION


Drag
negative pressure
Lift
Actual force produced by rotor
Wind
positive pressure
Actual force produced by rotor
Thrust
Wind
Side force

[122]

controlled automatically to maintain a constant ship speed. When the sail thrust is more than a certain value, the main engine output cannot be made less than the lower limit of service power, so that the ship speed becomes higher than the set speed, as shown on Figure 9.10 [123, 28].

- MINI LACE, a 3,100 dwt general cargo vessel operates with 270 m^2 cat rig between Jamaica and New Orleans. The sail, made of dacron, is electrically operated from the bridge. The sail system provides hydraulically powered control of the amount of exposed sail; the angle of the sail in relation to the vessel, and clew outhaul and downhaul tension. Since the installation of the sail power unit, the ship bunkering records show that the projected fuel savings of 20% (on which the decision of retrofit was made) has been exceeded by a substantial margin. The calculated range of rig payback, depending on ship route scenarios is 1.7 to 4.4 years [28].

  Based on owner's records for fuel consumption the average annual savings reached $48,000 for the current fuel price of U.S. $327 per tonne. The increased revenue from speed increase and the consequent extra voyages was $9,200 for a time charter of $2,000 per day. The total yearly benefit was calculated as equal to $57,200. When the investment period was based simply on rig cost and first year's fuel savings, the rig cost would return in 4.4 years. However, if the same criterion is applied to a year of operation on favourable sailing routes the payback period would decrease to 1.7 years [28].

  The ship's performance records shown in Figure 9.11 are a good illustration of her sail advantages.

9.7 Nuclear Energy

The use of nuclear energy may have a significant potential for marine propulsion. To date nuclear propulsion has been used mainly in Navies and in some experimental merchant vessels: SAVANNAH, OTTO HAHN, MUTSU, and the Soviet icebreakers LEONID BREZHNEV, LENIN, ARKTICA, and SIBIR.

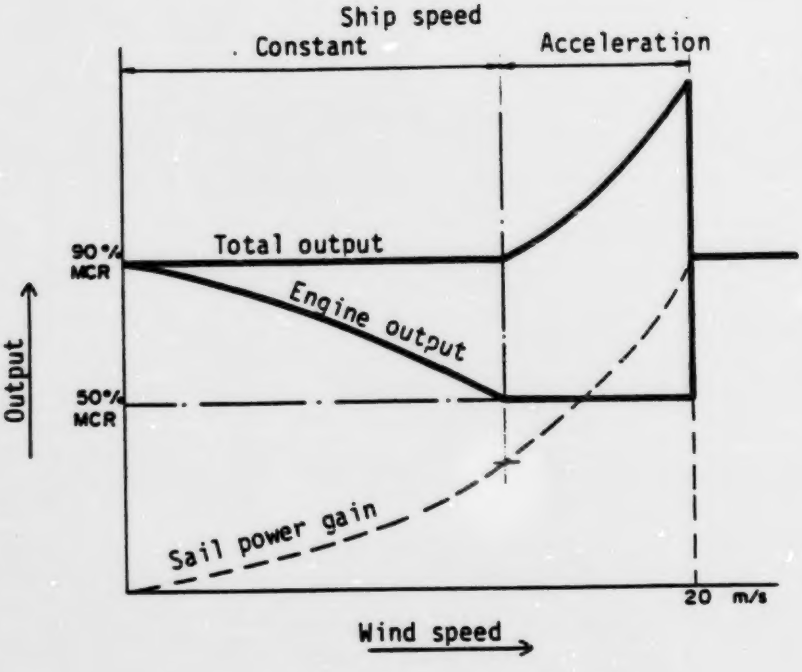
The experience gained during the operation of these ships has proved that nuclear fueled boilers at sea are practical, safe and can be used by commercial ships. However, due to economical, political, social, and environmental rather than to technological reasons, nuclear propulsion has not yet been used in any pure commercial vessel.

It is believed that as the cost of hydrocarbon fuel increases, or its availability becomes less certain, nuclear propulsion will be a seriously alternative which will be considered. Some recently published studies claim that the economics of nuclear energy may be found attractive for ships for the 1990's.

At present, the cost of uranium, which is utilized as a fuel in nuclear ships, is significantly higher on a per kW basis, than fossil fuel.

FIGURE 9.10

ENGINE OUTPUT CONTROL
M/V "SHIN AITOKU MARU"

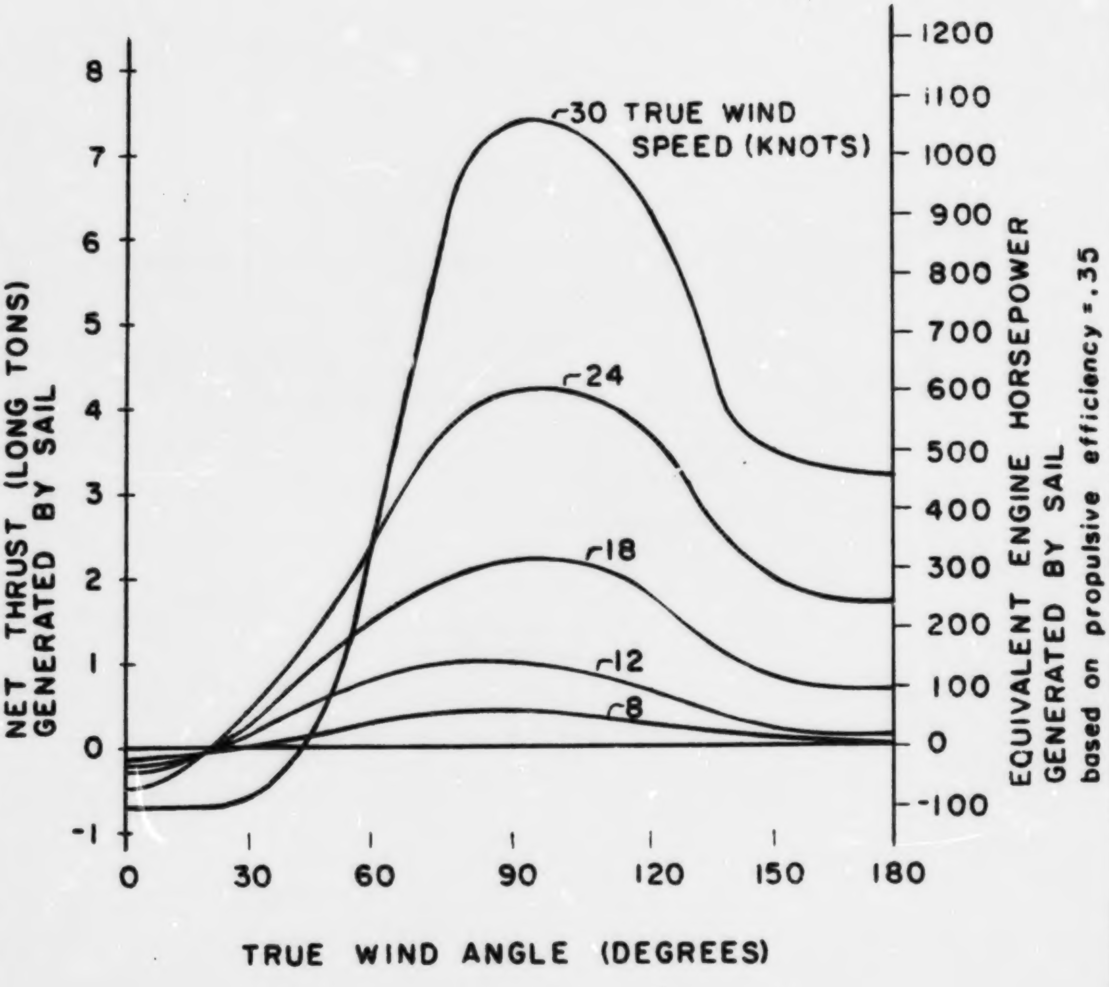


Engine Output Control

[28]

FIGURE 9.11

RIG HORSEPOWER M/V "MINI-LACE"



[28]

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However, with each rise in fuel oil price nuclear power will become more economical.

The economic case for nuclear propulsion can be more clearly estimated now that fuel oil costs have increased. Before the energy crisis, in 1973, high powered vessels of 90,000 kW were considered as the most suitable for nuclear propulsion. Burning oil in these high powered vessels represents a prohibitive operating cost, but a nuclear powered ship of this type would today be 12% more economic with respect to fuel costs [115].

All nuclear merchant ships' reactors designed since the SAVANNAH, have utilized low enrichment uranium as fuel. The availability of uranium in the future is almost as problematic as the supply of oil fuel. Several observers have come to the conclusion that the supplies of nuclear fission material are likely to be exhausted possibly before supplies of oil. The availability of uranium fuel can be extended by its regeneration from spent fuel in fast breeder reactors. However only experimental fast breeder reactors are presently available and the future development of these reactors is debateable [125].

Nuclear energy would be economically feasible in merchant ship application above a certain power level. The initial and operating costs of the nuclear vessels with the lower propulsion power in the range of the existing plants are higher than those of conventional systems. Studies carried out in the United Kingdom [122] made a cost comparison of the four types of vessels: conventional, rotor ship (equipped with Flettner rotors), coal fired ship and nuclear ship. The rotor ship was chosen as the representative of sail assisted ships due to the low initial cost of the rotors. For the coal fired ship a newly built vessel with fluidised bed combustion system was chosen, while for nuclear ship the vessel incorporating an integral pressurized water reactor of 7,500 kW was used. The results are shown on Table 9.4.

Another possibility, for the more distant future, possibly around the year 2000, is the use of nuclear fusion. This is in a very early stage of development and the possible techniques and costs are most uncertain. The materials for fusion; isotopes of hydrogen and the metal lithium, are comparatively easily obtained. The main problem with nuclear fusion is the enormous temperatures of around 45 million K degrees minimum which are produced in the process and the difficulty of containing matter at that temperature. One system put forward is a reactor in which pellets containing deuterium and tritium are struck by laser beams to induce fusion. This might enable a reactor of moderate size suitable for smaller ships to be produced in the future [125].

One more indirect application of nuclear energy for marine propulsion purposes is the production of hydrogen. Very large scale shore based reactors might be used to generate hydrogen which could then be converted to fuels suitable for marine use. This use of nuclear power would leave the actual machinery on board the ship looking similar to today.

TABLE 9.4

COST COMPARISON FOR VARIOUS PROPULSIVE SYSTEMS (1979 PRICES)

ROUTE A (Southampton - New York - Southampton)

SHIP TYPE	OPERATING COST (U.S.$)	FUEL COST (U.S.$)	TOTAL COST (U.S.$)
Conventional	154,460	59,680	214,140
Rotor Ship	175,320	45,060	220,380
Coal Ship	168,180	72,120	240,300
Nuclear Ship	368,540	23,780	392,320

ROUTE B (Southampton - Fremantle - Southampton)

SHIP TYPE	OPERATING COST (U.S.$)	FUEL COST (U.S.$)	TOTAL COST (U.S.$)
Conventional	505,000	166,040	671,040
Rotor Ship	621,880	116,600	738,480
Coal Ship	549,800	164,260	714,060
Nuclear Ship	1,191,580	66,200	1,257,780

[122]

9.8 Wave Energy

Utilization of the energy of sea surface wave motion for ship propulsion can be obtained theoretically, directly through a special device fitted on board the ship or indirectly through storing wave energy ashore in another form and then supplying it to the ship.

Successful tests were carried out recently in Norway involving the use of wave energy to propel boats. The system consists of a moveable wing placed horizontally on an axis beneath the boat. The wing moves up and down in tune with the movement of the boat in the waves. First indications are that wave power will be able to propel boats up to about 50 meters long and will be more effective when the boat is sailing against the direction of the waves than with them [126].

The use of wave energy to propel ocean going vessels is not very likely.

9.9 Solar Energy

There are two known methods of converting solar energy to a form which allows its application to power systems.

Firstly, solar energy can be concentrated at either a central point or axis by means of reflecting surfaces. The energy can then be absorbed and collected in fluids within systems of plate collectors, tubes and channels. Secondly, solar energy can be directly converted to an electrical current by banks of photovoltaic cells.

The output of solar energy absorbing systems is limited according to variations of latitude, time of year, and time of day. The quantity of energy collectad also depends upon the effective area of the absorbing surface. Reflecting systems would require constant tracking to maintain focus with normal ship motion; absorption systems would produce a small amount of heat which could be difficult to ulitize economically.

Photovoltaic cells, therefore, seem to give the best prospects for ship application in the future. The very low efficiency of the cells would require substantial numbers to be employed and cause such designs to be very expensive. However, the development of this technology could change the cost of solar energy considerably in the future.

9.10 Ocean Thermal Energy

Ocean thermal energy could be utilized to make an alternate fuel such as hydrogen or hydrogen carriers.

Sea thermal power plants have been proposed which use temperature differences in the ocean to produce electrical power. The temperature gradient between the warm surface water and the colder water at a depth of 600 - 1,500 metres may vary from as little as 18°C to approximately 30°C depending on the latitude.

The basic principles of operation are similar to steam power plant operation and have either an open or closed cycle approach. In a closed cycle, the warm surface water is pumped down to a heat exchanger where a suitable liquid (i.e., propane or ammonia) is turned into a pressurised vapour by heat transfer and then expands though a turbine to drive a generator. The exhaust from the turbine goes to the condenser which is cooled by cold water from deep in the ocean. Figure 9.12 shows the principles of operation of such a plant [127].

9.11 Ocean and River Currents' Energy

Similar to the sources of energy discussed previously, the hydraulic energy contained in ocean and river currents could be recovered to make alternative fuels for ships. Some research is under way to determine the economic feasibility of the various systems.

9.12 Summary and Comparison of the Alternative Fuels and Energy Sources

A list of energy sources and alternative fuels which have potential for marine application has been published by the U.S. Maritime Transportation Research Board of the National Academy of Science [113]. Table 9.5 presents this list of twenty energy sources along with a qualitative ranking for each source. The table needs some explanation:

- Direct use applies to using that particular energy source as the means to propel the vessel, indirect means that the energy source could be used to provide hydrogen or the various hydrogen carrier fuels.

- Level of technological development
  - High - Commercial State
  - Medium - Demonstration (Pilot) State
  - Low - Research State

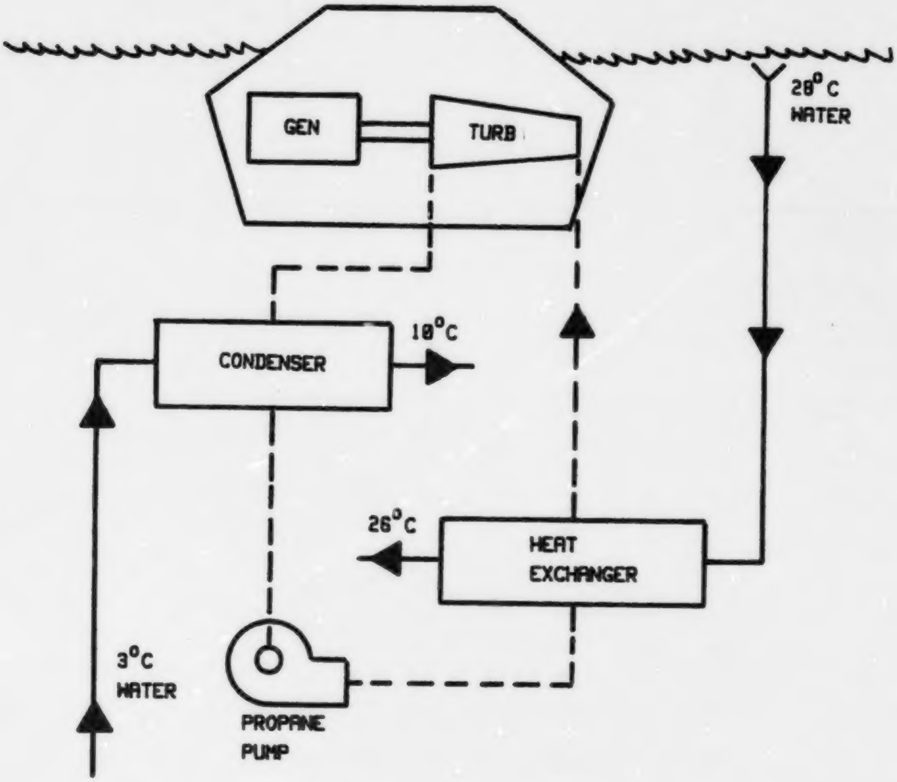
- Present availability for use as a fuel
  - High - Widely Available
  - Medium - Available in Limited Quantities
  - Low - Not commercially Available

- Safety in shipboard handling
  - Equal - At least equal to the safety of residual fuel oil
  - Low - Less safe than residual fuel oil

- Probability of economic production
  - High - High probability of being produced at competitive costs
  - Medium - Marginal probability of being produced at competitive costs
  - Low - Not likely to be produced at competitive costs

FIGURE 9.12

OCEAN THERMAL ENERGY CONVERSION PLANT


GEN
TURB
28°C WATER
CONDENSER
10°C
26°C
HEAT EXCHANGER
3°C WATER
PROPANE PUMP

[127]

TABLE 9.5

QUALITATIVE RANKING OF CANDIDATE ALTERNATE FUELS AND ENERGY SOURCES

	LEVEL OF TECHNOLOGICAL DEVELOPMENT	PRESENT AVAILABILITY FOR USE AS A FUEL	LOWER HEATING VALUE kJ/kg	SAFETY IN SHIPBOARD HANDLING	SAFETY TO PERSONNEL	PROBABILITY OF ECONOMIC PRODUCTION	POTENTIAL MARKET SIZE FOR MARINE PROPULSION SYSTEMS	TYPES OF PROPULSION SYSTEMS READILY ADAPTABLE TO FUEL BY YEAR 2000**	ADAPTABILITY TO EXISTING FUEL STORAGE SYSTEM	EXISTING SHIPS BY YEAR 2000	FUTURE SHIPS BY YEAR 2000	TIME FRAME
1. Synthetic Liquid Fuel Oil from Coal	Medium	Low	40,300	Equal	Less	High	High	1, 2, 3, 4	High	High	High	90's
2. Synthetic Liquid Fuels from Shale Rock	Medium	Low	42,300	Equal	Less	High	High	1, 2, 3, 4	High	High	High	80's
3. Synthetic Liquid Fuels from Tar Sands	High	Medium	(42,300)	Equal	Equal	High	High	1, 2, 3, 4	High	High	High	80's
4. Methyl Alcohol	High	Low	20,100	Less	Less	High	Medium	1, 3, 4, 5	Medium	Low	Low	90's
5. Methanol/Coal Slurries (40/60)	Medium	Low	23,300 26,000	Less	Less	High	Medium	4	Low	Low	Low	2000+
6. Ethyl Alcohol (Ethanol)	High	Medium	26,900	Less	Less*	Low	Medium	1, 3, 4, 5	Low	Low	Low	Present
7. Gasoline/Alcohol Blends (90/10)	High	Medium	(41,400)	Less	Less	Medium	Low	1, 3, 4, 5,	Low	Low	Low	Present
8. Pulverized Coal/Oil Slurries (40/60)	Medium	Low	36,700 39,500	Equal	Equal	High	High	2, 4	High	Medium	High	80's
9. Hydrogen	High	Low	120,000	Less	Less	Medium	Low	-	-	-	-	2000+
10. Ammonia	High	Medium	18,700	Less	Less	Medium	Low	-	-	-	-	2000+
11. Hydrazine	High	Low	17,000	Less	Less	Low	Low	-	-	-	-	2000+
12. Methane	High	High	50,000	Less	Less	High	Low	1, 3, 4, 5	Low	Low	Low	80's
13. Nuclear (D)(1) (I)(2)	High Medium	High Low	N.A. N.A.	N.A. Varies	N.A. Varies	High Varies	Low Low	1, 4 Varies	N.A. Varies	Low Varies	Medium Varies	Present 2000+
14. Coal	High	High	27,900 34,900	Equal	Equal	High	High	4	Low	Low	High	Present
15. Wood (Air Dried)	Medium	Medium	12,800 18,600	Equal	Equal	High	Low	4	Low	Low	Low	80's
16. Solar Energy (D) (I)	Low Low	Low Low	N.A. Varies	N.A. Varies	N.A. Varies	N.A. Varies	N.A. Low	N.A. -	N.A. -	- -	- -	N.A. 2000+
17. Wave Energy (D) (I)	Low Low	Low Low	N.A. Varies	N.A. Varies	N.A. Varies	N.A. Varies	N.A. Low	N.A. -	N.A. -	- -	- -	N.A. 2000+
18. Wind Energy (D) (I)	High Medium	High Low	N.A. Varies	.N.A. Varies	N.A. Varies	Low Varies	Low Low	N.A. -	N.A. -	Low -	Medium -	Present 2000+
19. Ocean Thermal (D) (I)	Low Low	Low Low	N.A. Varies	N.A. Varies	N.A. Varies	N.A. Varies	N.A. Low	N.A. -	N.A. -	- -	- -	N.A. 2000+
20. Ocean Current (D) (I)	Low Low	Low Low	N.A. Varies	N.A. Varies	N.A. Varies	N.A. Low	N.A. Low	N.A. -	N.A. -	- -	- -	N.A. 2000+

(1) D = Direct; (2) I = Indirect; 1-Gas Turbine; 2-Diesel and Free Piston; 3-Internal Combustion Piston Engine; 4-Steam; 5-Fuel Cell; N.A.-Not Applicable; ( )-Denotes Estimated Values; *-Denatured; **-Refer to MTRB Study; -Possibility of Adoption by Year 2000 too Remote to Consider.

- Potential market size for marine propulsion systems
  - High - Large number of vessels might adopt fuel
  - Medium - Moderate number of vessels might adopt fuel
  - Low - Few if any vessels would adopt fuel

- Adaptability to existing fuel storage sytems takes into consideration acceptability by Regulatory Agencies and Classification Societies.

  - High - Adaptable with minor or no modifications
  - Medium - Adaptable only after major modifications
  - Low - Adaptation of fuel might require such extensive modifications of vessel or fuel handling systems as to make the change economically unattractive

- Overall ranking. Takes into consideration all the categories used in comparing the fuels for both existing and new construction.

- Time frame. Estimate of decade in which each fuel will become available in commercial quantities.

To evaluate the costs of the alternative fuels, Figure 9.13 has been prepared. These costs are based on data from the U.S.M.T.R.B. study and from other literature [34, 113, 127]. The quoted costs are those for manufacturing fuels, to which should be added costs of storage, transportation and distribution. These additional costs, due to their high uncertainty, have not been assessed.

In the case of hydrogen, methane and natural gas the total cost of energy from these sources should be increased by the liquefaction cost since they will be used mostly in liquid form.

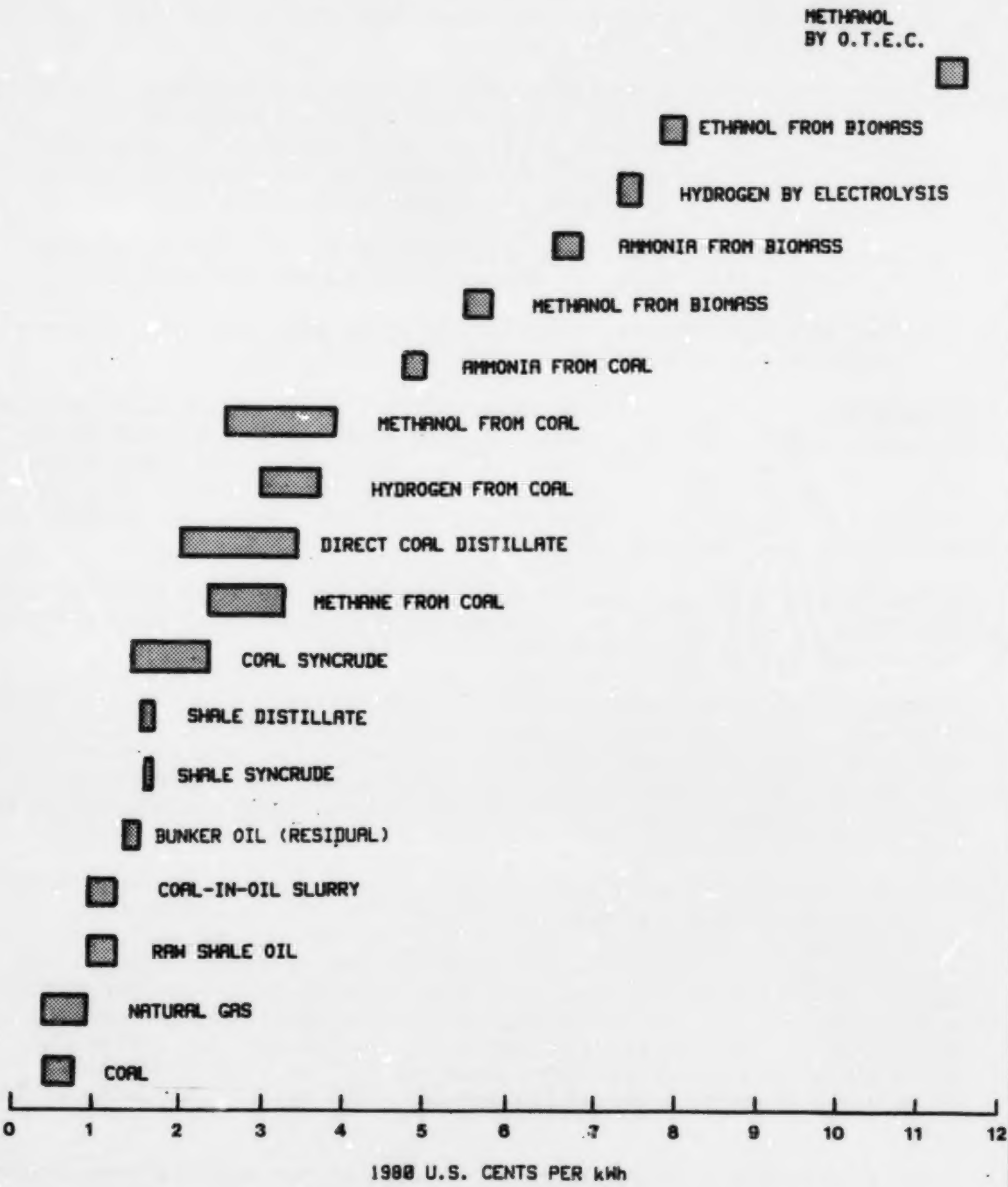
Another suitable method to compare alternative fuels is by thermal efficiency, i.e., the ratio of the obtained shaft horsepower to the required amount of energy contained in a basic resource. This measure of efficiency includes all energy lost between the resource and the engine output, e.g., conversion to fuel losses, transportation losses, losses in the engine etc. The losses for some fuels are referred to in Figure 9.14 [34, 113,127].

Another consideration which must be taken into account to enable alternative fuels to be graded is that of "impact on the environment". It is particularly important in the case of such inland waterway vessels as lakers that power plant emissions should meet air quality regulations. Since the amount and quality of the emission depends on many factors of the combustion process as well as the efficiency of the emission control devices, the estimating of the emission effects on environment is difficult. The U.S.M.T.R.B. study made estimates based on fuel properties and known types of emission in certain combustion systems. The results are shown in Table 9.6.

Finally, Figure 9.15 shows the relative costs of the electric power produced from the different sources of energy. The cost of the power depends not

FIGURE 9.13

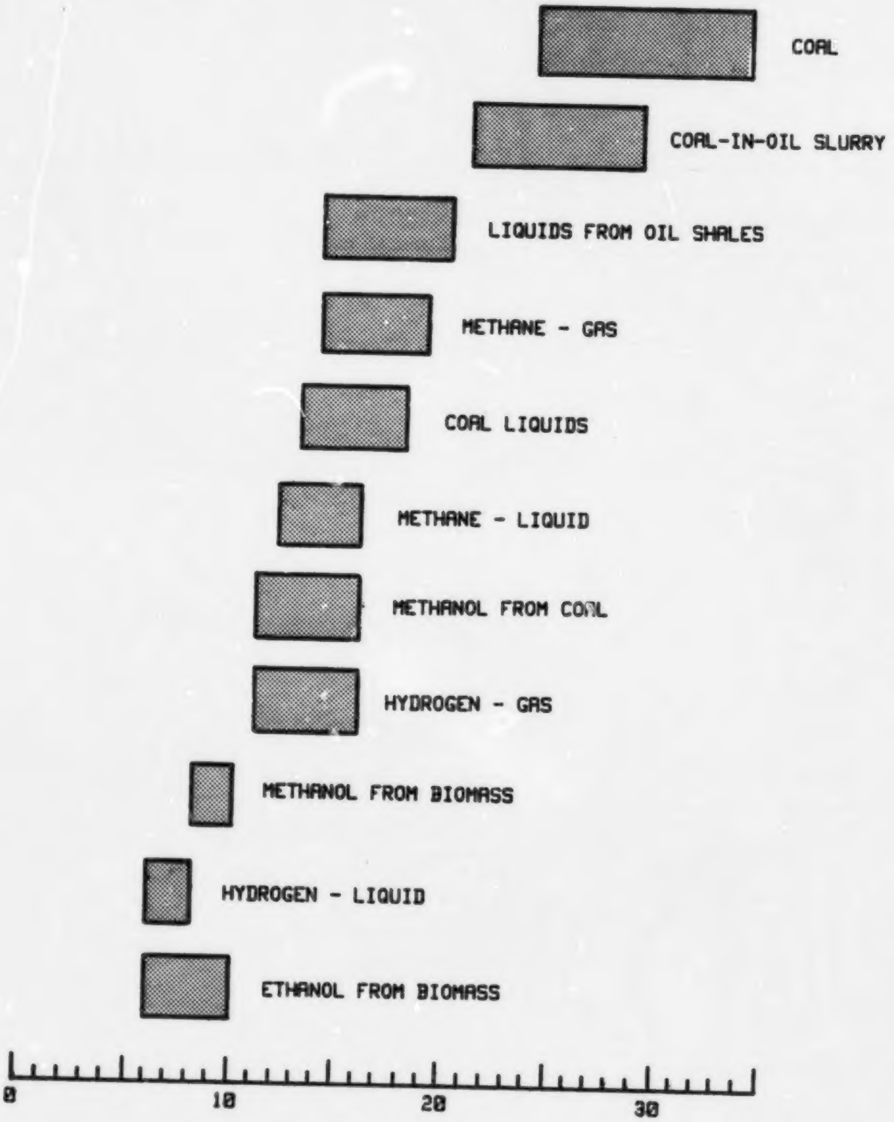
COST COMPARISON FOR THE PRODUCTION OF ALTERNATIVE FUELS



[34] [113] [127]

FIGURE 9.14

OVERALL THERMAL EFFICIENCY (MINE TO STEAM TURBINE SHAFT HORSEPOWER)



% OVERALL THERMAL EFFICIENCY

[34] [113] [127]

TABLE 9.6

EMISSIONS EFFECTS OF FUELS AND PROPULSION PLANT - IN USE

	OTTO CYCLE	DIESEL	GAS TURBINE		
FUEL	NORMAL SPARK IGNITION	(A)	STATIONARY	AIRCRAFT	STEAM
Petroleum - Present Specification	0	0	0	0	0
Coal	NA	NA	-2	NA	-2
Coal/Methanol Slurry	NA	-2	-2	NA	-2
Coal/Oil Slurry	NA	-2	-2	NA	-2
Coal Liquids	0 to -1	0 to -1	-1 to +1	-1	0 to -1
Shale Oil Liquids	-1	0	-1	-1	0
Methanol	+1	+1	+1	+1	+1
Hydrogen	+2	+2	+2	+2	+2
Ammonia	-1	-1	-1	-1	-1
Ethanol	0	+1	+1	0	0
Methanol	+1	+1	+2	+1	+2
Hydrazine	NA	0	0	0	0

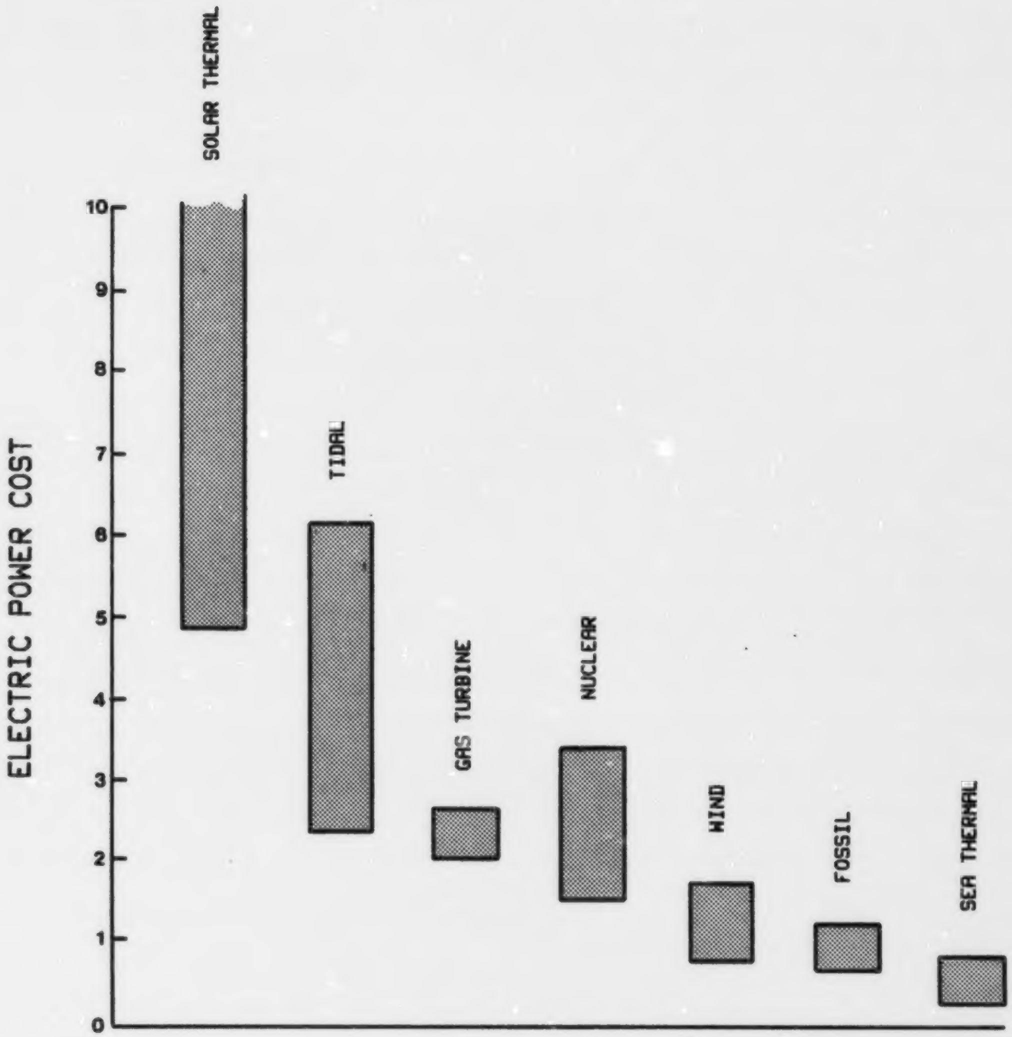
Code: +2 = Better in emissions of at least two pollutants.
+1 = Better in emissions of one pollutant.
0 = Comparable to current fuel.
-1 = Poorer than conventional fuel.
-2 = Much poorer than conventional fuel.

(A) Considers possible use of a dual fuel system.

[113]

FIGURE 9.15

RELATIVE COSTS OF ELECTRIC POWER
FROM VARIOUS ENERGY SOURCES



only on the plant cost but also on how much of the time the power can be produced. This has been taken into account in this figure. Though the figure shows electric power costs, it can also be an illustration of the possible cost of power plants producing energy for direct ship propulsion.

10.0 FUTURE PROPULSION SYSTEMS AND EQUIPMENT

10.1 General

The intent of this section is to discuss power plants which may be suitable for marine propulsion in the near or long term. Some of the designs which are described have begun to be applied to existing ships.

As a rule, almost all future propulsion systems will utilize alternative fuels to oil. This section can be considered as an illustration of some of the possible applications of the alternative fuels which have been previously discussed.

10.2 Coal Fired Systems

The only technology presently available for utilization of coal onboard ship is coal fired boilers. Ships fitted with steam boilers using coal as fuel to supply steam to either reciprocating or turbine engines were in common use up to the early 1930's, when oil bunkers become more popular. Coal handling technology and the generation of steam by coal fired boilers have made rapid advances in the past thirty years in the land based electrical utilities industry. Modern coal fired boilers and coal handling systems are considerably more sophisticated than their predecessors of thirty years ago.

There are four principal methods of burning coal in boilers:

- Oil or water based coal slurry firing
- Pulverized coal firing
- Mechanical stoker firing
- Fluidized bed combustion

10.2.1 Oil or Water Based Coal Slurry Firing

Slurry firing has the advantage of ease of handling of the fuel and simple application to existing ship's boilers. There are difficulties in the production of stable slurries which make this fuel too expensive at this time.

10.2.2 Pulverized Coal Firing

There are two principal types of pulverized coal firing systems - the direct system, and the indirect or bin system. In the direct system, lump coal is fed through a dryer (hot air at 315°C) into a table or ball mill pulverizer where it is ground to fine mesh. The pulverized coal is transported by air to the fuel nozzles and into the furnace. In the indirect system coal is pulverized and stored in gas inerted bunkers. The pulverized coal is transported from the bunker tanks directly to the fuel nozzles of the boiler.

A proposal to use the bin system utilizing combustion gas from the boiler as a means of inerting the mill and transporting the coal to a ready use storage bunker, is shown in Figure 10.1 [129]. This system still has the disadvantage of a requirement for mills on board ship and significant auxiliary power necessary for fuel preparation, equal to 10 - 15 kW/tonne of coal per hour.

Pulverized coal firing of boilers is widely used ashore and is more efficient than stoker fired units. However, the equipment required to pulverize the coal is heavy, expensive and is prone to failures. In land practice pulverizer mills are not reliable and often more than one extra mill is fitted to ensure full load operation can be achieved as required. The efficiency of pulverized coal burning boilers is about 88% whereas the stoker fired boiler efficiency is about 84%. Pulverized fuel boilers are best suited for capacities of above 150 to 200 tonnes per hour steam generation which is considerably in excess of the requirement for most merchant ships.

10.2.3 Mechanical Stoker Firing

The mechanical stoker is the best system available at present for coal firing onboard ship. The spreader stoker with continuous ash discharge (or travelling) grate can utilize efficiently a range of different coals and gives the system a rapid load change characteristic. The stoker requires a relatively coarse fuel mixture, and fuel preparation equipment onboard ship may, therefore, be a simple grader and crusher. Figure 10.2 illustrates a modern marine stoker fired boiler fitted with a continuous ash discharge type grate.

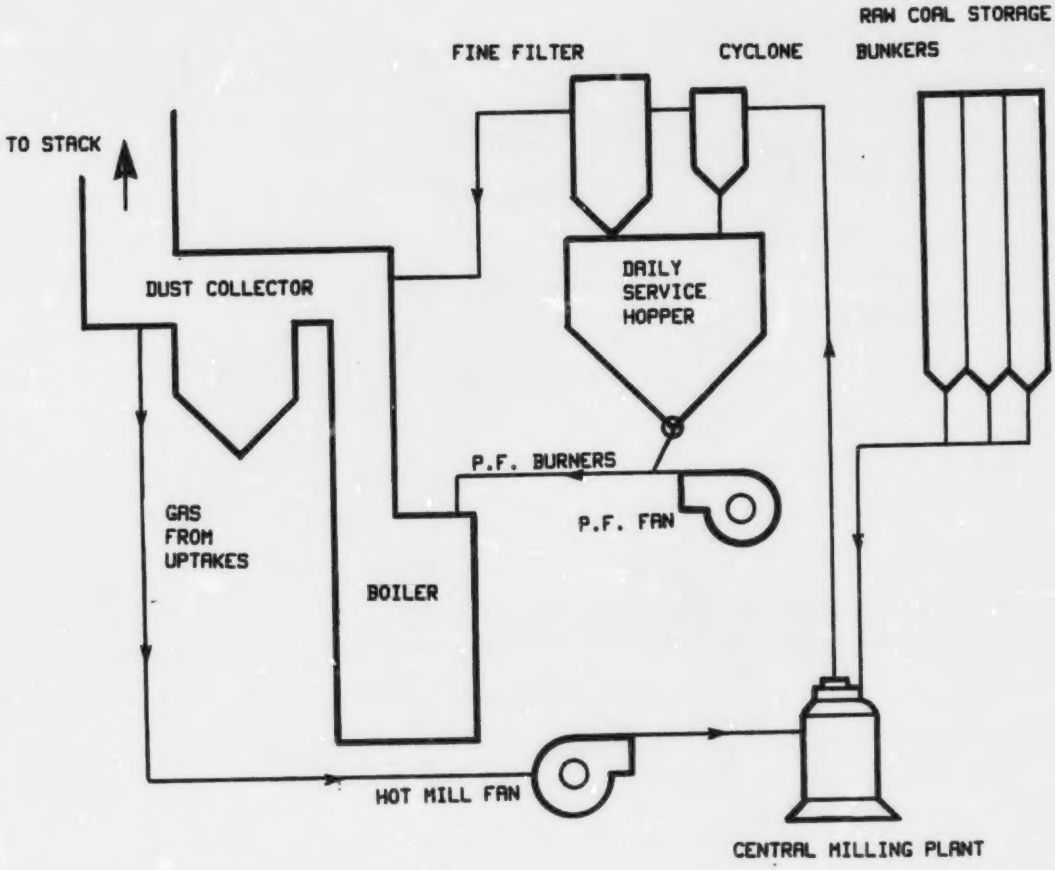
Fuel is distributed across the grate by a mechanical spreader, which is adjusted for optimum coal distribution. Integrated into the spreader is a chain feeder which regulates the coal flow in the same manner as the fuel oil regulating valve in an oil fired system. Figure 10.2 also shows fuel rates necessary to produce power of 15,000 kW for some typical U.S. coals and illustrates the difference of coal feed rate between fuels of different calorific value. As an example, the firing rate and thus the grate speed is nearly twice for Northern Plains Lignite as compared to Eastern Bituminous coals [130].

The grate of the boiler is essentially a slow moving endless chain link bed which can be adjusted to travel at different speeds. A typical operating speed is 5 meters per hour. The stoker spreads coal uniformly on the grate as needed to maintain boiler pressure and steam flow. The quantity of fuel on the grate depends upon boiler load and coal type and may be regulated as required and according to the type characteristics of the coal to be burnt. Since the recommended coal mixtures contain approximately 50% of smaller particles, ignition is rapid. Boiler response time is not as rapid as for oil or pulverized coal fired boilers, but is sufficient for marine use. A steam dump system from the boiler directly to the condenser is generally fitted to improve the rapidity of load change.

Air is supplied both below and above the grate. Approximately 80% of the combustion air is admitted to the interior part of the grate by means of windboxes located on each side of the grate. The air passes through the perforations in the grate for grate cooling purposes as well as providing air for the combustion.

FIGURE 10.1

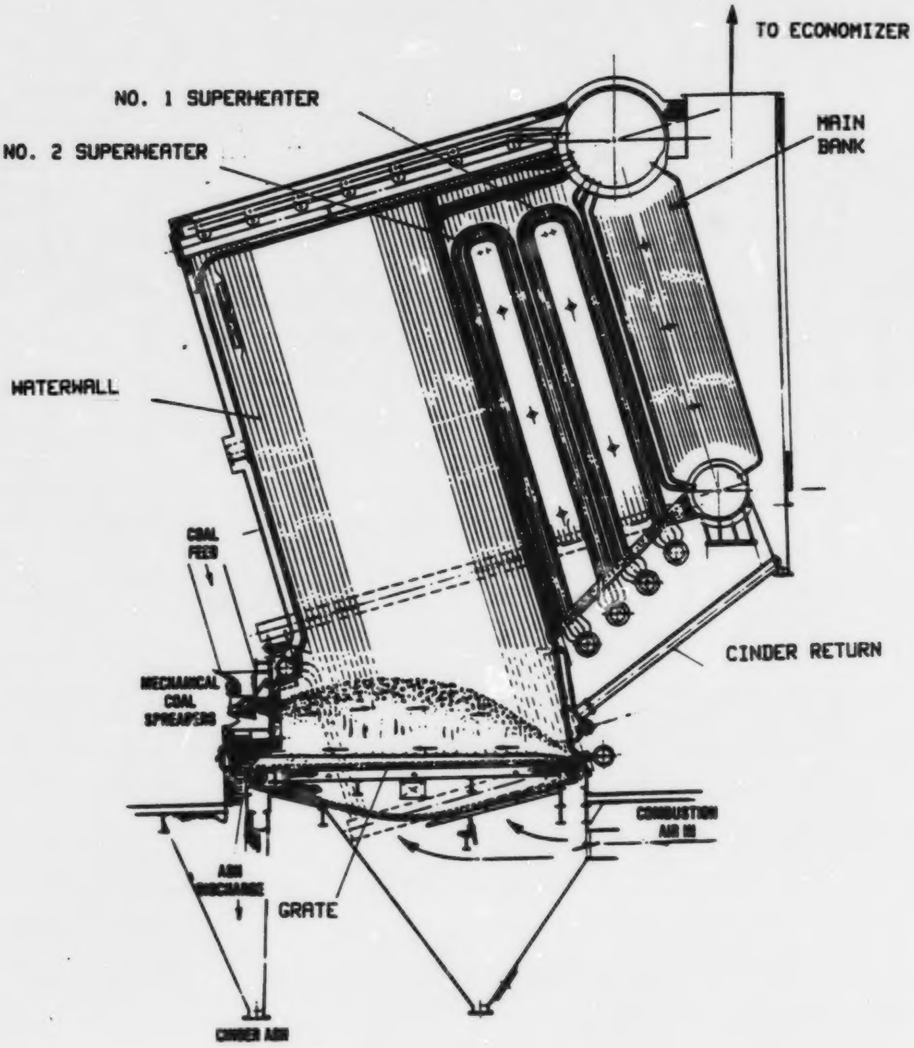
BIN AND FEEDER SYSTEM FOR PULVERIZED-FUEL


RAW COAL STORAGE
BUNKERS
FINE FILTER
CYCLONE
TO STACK
DAILY
SERVICE
HOPPER
DUST COLLECTOR
P.F. BURNERS
P.F. FAN
GAS
FROM
UPTAKES
BOILER
HOT MILL FAN
CENTRAL MILLING PLANT

[129]

FIGURE 10.2

STOKER FIRED MARINE BOILER


TO ECONOMIZER
NO. 1 SUPERHEATER
NO. 2 SUPERHEATER
MAIN
BANK
WATERWALL
COAL
FEED
MECHANICAL
COAL
SPREADERS
CINDER RETURN
COMBUSTION
AIR IN
ASH
DISCHARGE
GRATE
CINDER ASH

	EASTERN BITUMINOUS	MIDWESTERN BITUMINOUS	SUB. BIT C	TEXAS LIGNITE	NORTHERN PLAINS LIGNITE
HCV (kJ/kg)	27,912	23,260	19,538	16,980	15,817
MOISTURE (%)	6	12	27	32	37
FUEL FIRED kg/hr	7,575	9,163	11,068	12,837	13,880

[128]

Additional air is supplied above the grate by means of nozzles which are served by a separate fan of non-preheated air. This air provides agitation to assure good mixing of the fines and gases, which are burning above the grate in the combustion air.

Coal burning boilers must have provisions to remove ash since the coal used for stoker firing may have an ash content of 20-25%. Approximately 70% of the ash falls off the end of the traveling grate below the spreader unit, the remainder drops into the hopper under the interior grate or is carried by flue gas to the upper parts of boiler where it is collected and removed.

Together with the ash, some unburned carbon is carried over by flue gas. The amount of carbon carry over depends upon the coal quality and the gas velocity through the boiler. Normally an arrangement for the recovery of this unburnt fuel for refiring is employed. However, there are practical and economical benefits of the extent of carbon particle recovery and usually 50-70% of the recovered cinder is injected again into the furnace.

A major problem connected with unburned particles is complying with numerous environmental restrictions on stack opacity when in port or steaming close to land. Particles in the uptake gases are removed by the use of mechanical or electrostatic types of dust collectors. The dust collectors are fitted after the economizer and before an induced draft fan. The complete dust collector consists of a number of collection tubes installed in a common casing. Depending upon the diameter and type of these tubes, collection efficiency of about 95% may be achieved. Two cyclone separators located in series could give an efficiency in excess of 98% [131].

The combustion control system of a stoker fired boiler is similar to that of an oil fired boiler. An example of a recent system design is shown in Figure 10.3 [132]. The coal fired ships built in Japan for Australian National Line (ANL) have been equipped with this system, developed by Mitsubishi. The system consists of the following subsystems for automatic control of boiler operations:

- Automatic combustion control including:
    - . coal feeder speed control
    - . grate stoker speed control
    - . under grate air control
    - . over fire air control

- Feed water control
- Steam temperature control
- Furnace draft control
- Smoke control

Besides these functions, Mitsubishi control system includes various boiler protection logic sequences for thorough reliability.

The ANL vessels can be considered as the standard against which future coal fired vessels will be built. These ships are fitted with a wholly automatic process of coal handling from bunkering to combustion in the boilers and is classed by Lloyd's Register of Shipping for UMS (Unattended Machinery Space) operation. Figure 10.4 shows the engine room arrangement.

FIGURE 10.3

CONTROL DIAGRAM FOR COAL FIRED BOILER

Control Diagram for Coal Fired Boiler

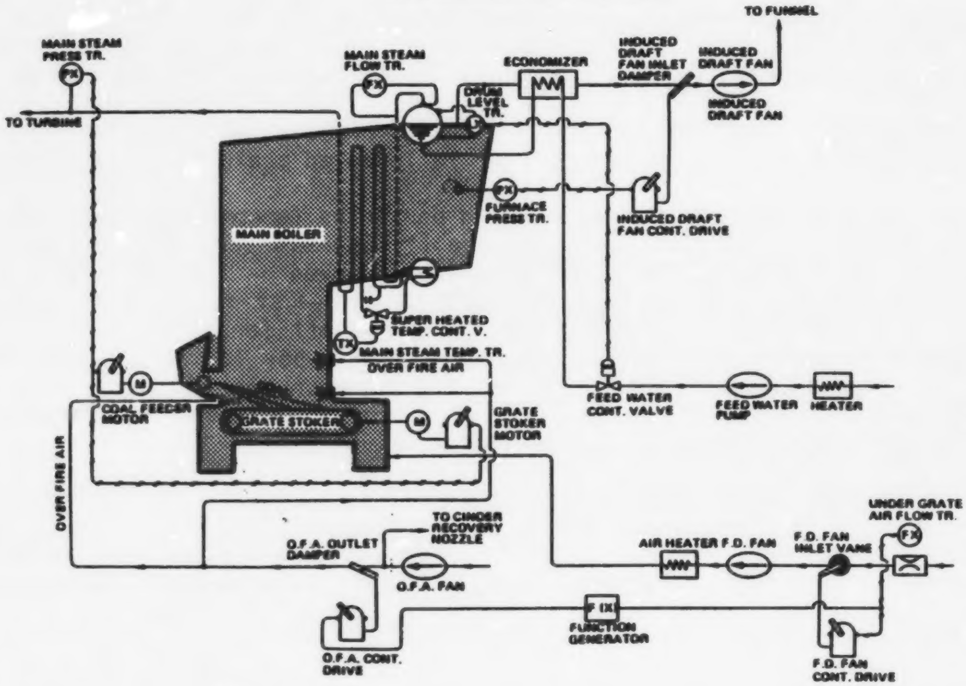
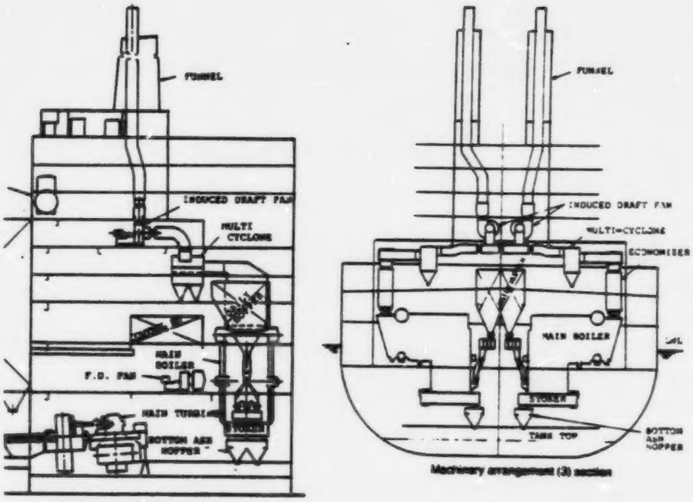

TO FUNNEL
MAIN STEAM PRESS TR.
FX
TO TURBINE
MAIN STEAM FLOW TR.
FX
ECONOMIZER
INDUCED DRAFT FAN INLET DAMPER
INDUCED DRAFT FAN
DRUM LEVEL TR.
INDUCED DRAFT FAN
FX
FURNACE PRESS TR.
INDUCED DRAFT FAN CONT. DRIVE
MAIN BOILER
SUPER HEATED TEMP. CONT. V.
TX
MAIN STEAM TEMP. TR.
OVER FIRE AIR
M
FEED WATER CONT. VALVE
FEED WATER PUMP
HEATER
COAL FEEDER MOTOR
GRATE STOKER
M
GRATE STOKER MOTOR
OVER FIRE AIR
UNDER GRATE AIR FLOW TR.
FX
TO CINDER RECOVERY NOZZLE
O.F.A. OUTLET DAMPER
O.F.A. FAN
AIR HEATER
F.D. FAN
F.D. FAN INLET VANE
F(X)
FUNCTION GENERATOR
O.F.A. CONT. DRIVE
F.D. FAN CONT. DRIVE

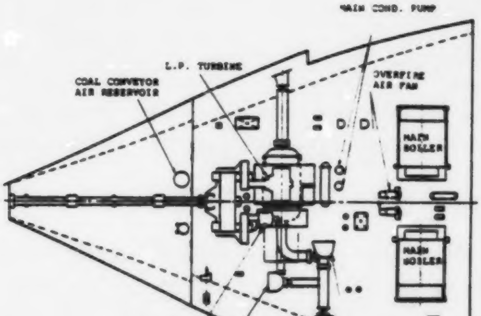
FIGURE 10.4

COAL FIRED MARINE STEAM PROPULSION PLANT


FUNNEL
INDUCED DRAFT FAN
MULTI CYCLONE
MAIN BOILER
F.D. FAN
MAIN TURBINE
BOTTOM ASH HOPPER
FUNNEL
INDUCED DRAFT FAN
MULTI-CYCLONE
ECONOMISER
MAIN BOILER
LWL
STOKER
TANK TOP
BOTTOM ASH HOPPER

Machinery arrangement (1) elevation

Machinery arrangement (3) section


MAIN COND. PUMP
L.P. TURBINE
COAL CONVEYOR AIR RESERVOIR
OVERFIRE AIR FAN
MAIN BOILER
MAIN BOILER
H.P. TURBINE
AUX. CIRCULATING PUMP
MAIN CIRC. PUMP
HIGH PRESS. WATER PUMP

Machinery arrangement (2) plan

MELVILLE SHIPPING LTD

The efficiency of a coal fired boiler is approximately 6% lower than a comparable oil fired boiler. This difference in efficiency is due to:

- 4.25% carbon loss, i.e., unburned carbon which is contained in the ash (this loss exceeds 8% if cinder reinjection is not installed)
- 1.35% moisture loss due to moisture in the coal
- 0.4% loss due to higher excess air requirement

The latest coal fired boiler designs are more efficient and these losses have been reduced. For instance Foster Wheeler claims that their stoker fired boiler E.S.D.C. (External Superheated "D" Type Coal Fired Boiler) has an 85% efficiency.

A comparison between an oil and coal fired boiler of Foster Wheeler design is shown in Table 10.1 [28].

Currently the capital cost of a coal fired ship is estimated at some 10 to 20% higher than that of a similar motor ship, depending on size and specifications. It may not be fair, however, to compare standard diesel engine plants with specially ordered coal fired steam plants at this early stage. The price of the coal fired propulsion system will be heavily influenced by the development cost and the difference can be narrowed by increased production of marine coal fired ships together with improved design [10].

Fuel cost savings will vary significantly with the relative cost between coal and oil. Figure 10.5 gives an example of the relationship between the magnitude of these savings against coal prices. The savings do not recognise such items as additional maintenance costs which may occur with coal fired systems.

The choice of the mechanical stoker type of coal fired boiler for modern merchant vessels reflects the first step in resurrecting the past marine experience before progressing to more modern technology as typified by pulverized fuel fired boilers and fluidized bed technology.

10.2.4 Fluidized Bed Combustion System (FBC)

Fluidized bed marine boilers are actively being developed and may be available for use in ships in the near future. This method has potential to utilize a wide range of coals as well as to achieve higher efficiency than stoker boilers.

The fluidized bed is not new technology. It has been used for a number of different purposes ashore for about 60 years, especially by the petrochemical industry. The fuel crisis has increased the interest in this system due mainly to the possibility of burning high sulphur, inexpensive coal and maintaining the content of the waste gases within the limits imposed by air pollution regulations on land. Today, fluidized bed combustion systems are being offered commercially and installed in many countries. For example, Foster Wheeler had constructed 34 FBC boilers for land application in the range 8,000 to 135,000 kg of steam per hour.

TABLE 10.1

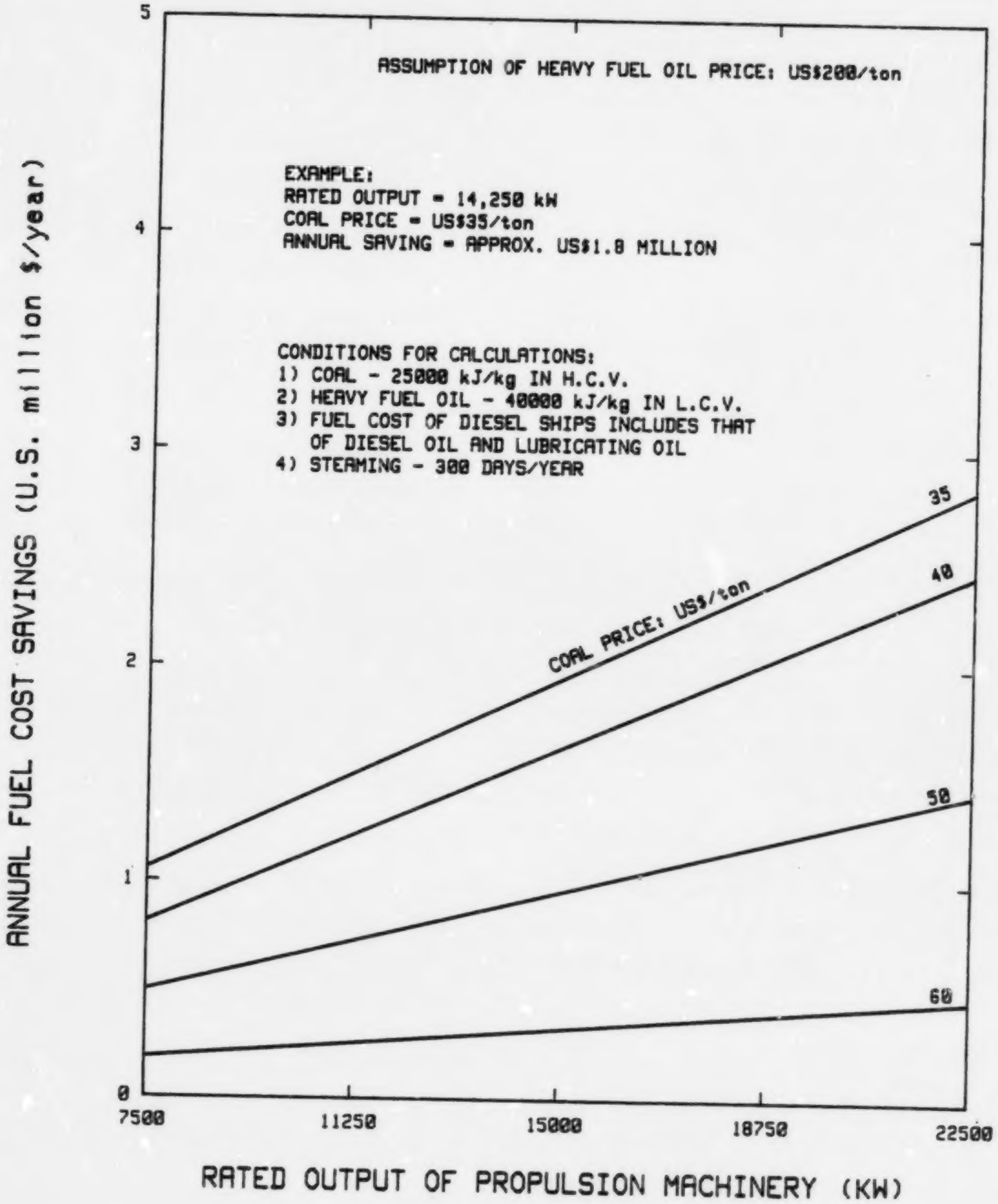
COMPARISON OF OIL AND COAL FIRED BOILERS

EVAPORATION	OIL FIRED NORMAL	COAL FIRED NORMAL
OUTPUT KGS PER HOUR	45,000	43,000
SUPERHEATER OUTLET PRESSURE KGS/CM^2	65	61
SUPERHEATER OUTLET TEMPERATURE °C	514	490
FEEDWATER TEMPERATURE °C	140	150
HIGHER CALORIFIC VALUE K.cal.Kg	10,280	5,344
AIR TO FUEL RATIO	15.2:1	9.5:1
EXCESS AIR PER CENT	10	35
EFFICIENCY PER CENT	88.5	85.0
FUEL RATE KG/HR	3,250	5,974
DESIGN AMBIENT AIR TEMPERATURE °C	38	38
AIR TEMPERATURE TO BOILER °C	125	120
FUNNEL TEMPERATURE °C	172	178
FURNACE VOLUME Cu.M.	46.2	140
HEAT RELEASE PER Cu.M. FURNACE VOLUME K.cal. HR.	723,500	228,072
WEIGHT OF BOILER (DRY) TONNES INCLUDING STOKER	131	250
WEIGHT OF ECONOMISER (DRY) TONNES	37	56
BASIC DIMENSIONS: PER BOILER		
WIDTH - ACROSS SHIP MM	7,400	8,400
FURNACE DEPTH - FORE AND AFT MM	3,590	4,900
HEIGHT ℄ STEAM DRUM TO BASE MM	7,900	14,650

[28]

FIGURE 10.5

ECONOMICAL EFFICIENCY OF COAL FIRED SHIP AS COMPARED WITH DIESEL SHIPS



ASSUMPTION OF HEAVY FUEL OIL PRICE: US$200/ton

EXAMPLE:
RATED OUTPUT = 14,250 kW
COAL PRICE = US$35/ton
ANNUAL SAVING = APPROX. US$1.8 MILLION

CONDITIONS FOR CALCULATIONS:
1) COAL - 25000 kJ/kg IN H.C.V.
2) HEAVY FUEL OIL - 40000 kJ/kg IN L.C.V.
3) FUEL COST OF DIESEL SHIPS INCLUDES THAT OF DIESEL OIL AND LUBRICATING OIL
4) STEAMING - 300 DAYS/YEAR

The principle of fluidized bed combustion is the burning of the combustible when suspended in the combustion air together with fine particles of sand, limestone or other suitable bed material related to the impurities in the fuel. The boiler has one or more fluidized compartment into which combustion air is introduced through nozzles at the bottom which imparts a swirling motion to the bed material. Steam or hot water tubes are immersed in this bed which contains material about 6 mm diameter which when agitated by the air, behaves like a fluid. The tubes absorb up to 50% of the heat output and thus keep the bed temperature low. Due to the swirling motion of the bed, heat transfer to the tubes is very high. The flue gases pass from the fluid bed to the convection part of the FBC as in a conventional boiler.

The combustion temperature of coal is between 800 to 900°C. At this temperature the sulphur present in the coal combines with the limestone in the bed material. The result is a dry, inert waste product, mainly gypsum, which can be disposed together with the ash. In this way 80 to 90% of the sulphur content in the coal is bound and not vented to the atmosphere. Furthermore, owing to the low combustion temperatures the NO_x emission are also greatly reduced. The coals should be in less than 50 mm diameter lumps when introduced into the bed.

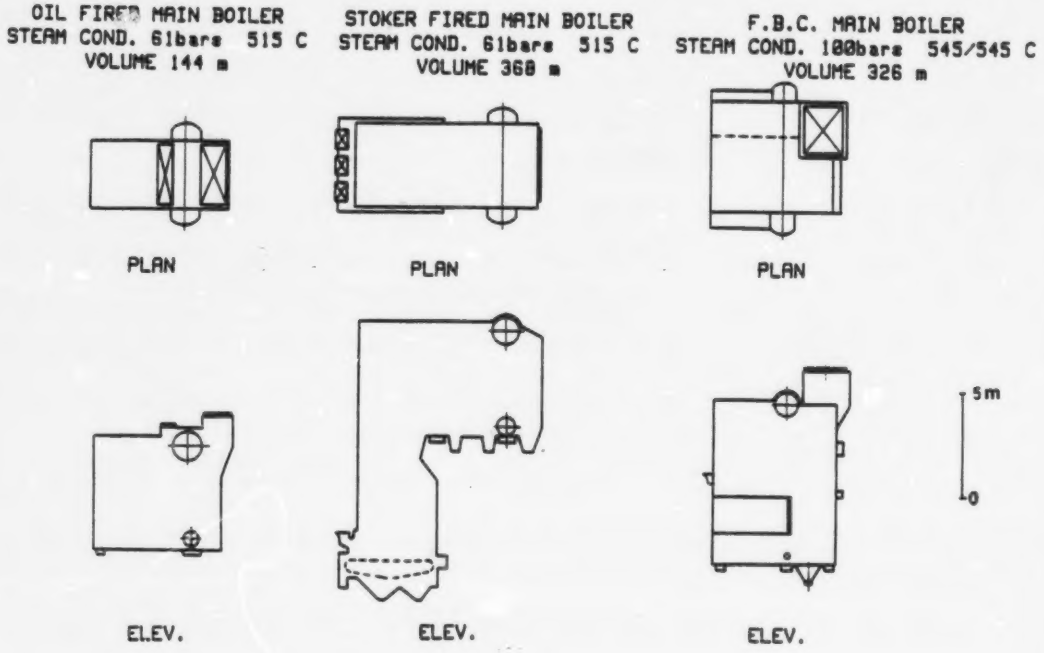
In terms of boiler efficiency, the FBC boiler does not represent a significant advance. Some improvements in efficiency are gained through the lower amount of excess air need compared to the spreader stoker fired boiler, 10% vs 30%, but the problem with carbon losses is mainly the same, i.e., a carbon recovery system is needed for achieving a reasonable efficiency. However, FBC boilers hold a potential for increasing the total cycle efficiency, since steam conditions can be increased. Due to the improved heat transfer conditions for the steam tubes submerged in the bed, steam temperatures above those obtainable with a stoker fired boiler may be employed. The improved heat transfer also reduces the size of superheaters and reheaters if needed.

The following advantages of the fluidized bed boiler are the main incentives for its use in marine application:

- The possibility for improvement of overall efficiency.
- Large flexibility in the coal quality.
- Smaller dimensions of the boiler due to the higher performance in the heat transfer, as shown on Figure 10.6 [133].
- Reduced fouling and corrosion of boiler surfaces.
- Faster response to changes in load condition than the conventional mechanical stoker boiler (but still not as fast as the oil fired boiler).
- Reduced SO_2, SO_3 and NO_x emission which can be important regarding any future emission regulations for coal fired ships.

FIGURE 10.6

DIMENSIONAL COMPARISON OF VARIOUS MARINE BOILERS


OIL FIRED MAIN BOILER
STEAM COND. 61bars 515 C
VOLUME 144 m
STOKER FIRED MAIN BOILER
STEAM COND. 61bars 515 C
VOLUME 368 m
F.B.C. MAIN BOILER
STEAM COND. 100bars 545/545 C
VOLUME 326 m
PLAN
PLAN
PLAN
ELEV.
ELEV.
ELEV.
5m
0

[133]

*
DESIGN POINT IS 14.7 MW

Examples of fluidized bed boilers design are shown on Figure 10.7. Figure 10.7 presents a schematic view of the Foster Wheeler boiler which utilizes a spreader stoker for coal supply to the bed [128].

The development of fluidized bed boilers, especially for marine application is inhibited by problems associated with the loss of bed material, unburned carbon particles in the gas stream and the loss of combustion efficiency when the bed inclines with ship's rolling and pitching.

The problem of excessive carry over of bed particles is chiefly overcome by the limitation of air speed through the bed. This speed should not exceed 2.4 m/sec, but in turn it should not be too low that the bed is not maintained in a fluid state. In order to improve the capability of capturing all the unburned carbon particles and to inhibit the loss of bed material, a recirculation system has been developed and applied in the power plants ashore. This system returns the ash and unburned carbon to the bed by means of special cyclone scrubbers downstream of the furnace. The recirculating equipment generally forms part of the combustion area of the boiler and is called Battel's system or Circulating Fluidized Bed system.

The problem due to ship motion has been investigated in some countries using models as well as full scale on board ship. The ship FJORDSHELL, owned by Shell International Marine Ltd., was fitted with an experimental waste heat boiler utilizing fluidized bed principles. The arrangement is shown in Figure 10.8 [136]. With several tonnes of red hot ash and sand within the boiler it is necessary to restrict the fluid bed motion. To limit the motion a system of baffles have been employed although swash plates within the bed would have achieved similar results. This experiment was reported to be successful although no other vessels, as far as is known, have been fitted with this type of boiler.

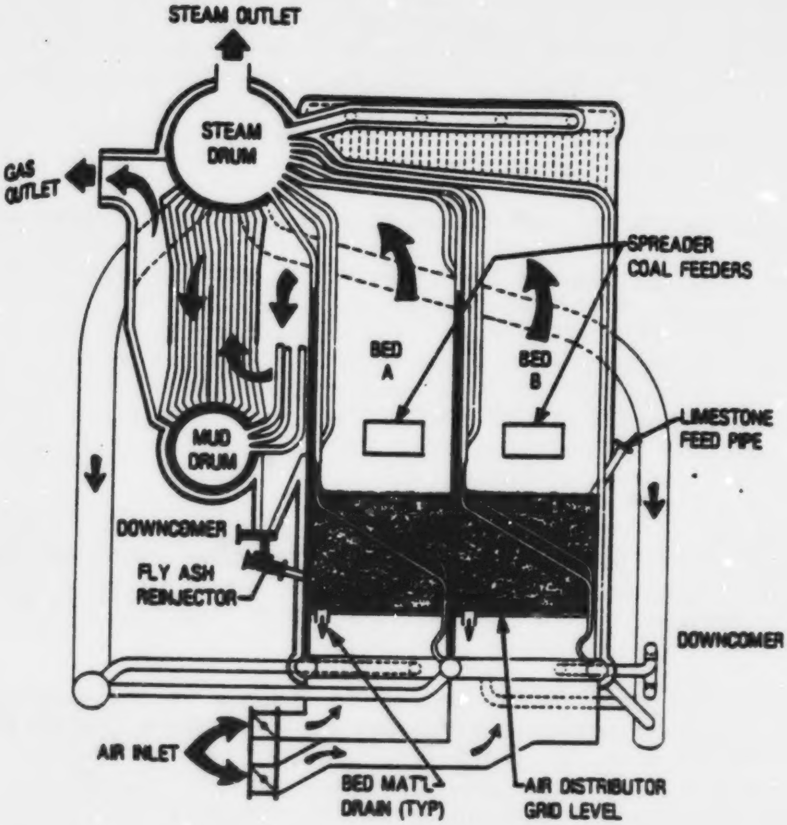
The operation of a fluidized bed boiler is not much more complicated than a mechanical stoker boiler. Start up requires some additional equipment which can be an in-bed or over-bed oil burner or a hot air generator which brings the bed temperature to the self ignition temperature of coal or the combustible to be used.

The dynamic response of the boiler to changes in the load is slower than that of a conventional oil fired boiler. Therefore controlability is less precise in the event of large load changes, if the coal feed rate and air flow rate are controlled only by signal of steam pressure. This can be overcome by excess steam dumping to the condenser combined with coal feed and air flow control. If a reheater is fitted, to avoid additional systems of cooling steam for reheater protection during astern operation, a controllable pitch propeller is recommended to allow the steam flow to be maintained through the reheater in all circumstances.

The fluidized bed boiler also presents a favourable feature for the improvement of plant efficiency at partial load. The superheated steam temperature shows a rising characteristic at part load and must be controlled. The reason for this is the position of the superheater installation in the fluidized bed. In conventional oil and stoker fired

FIGURE 10.7

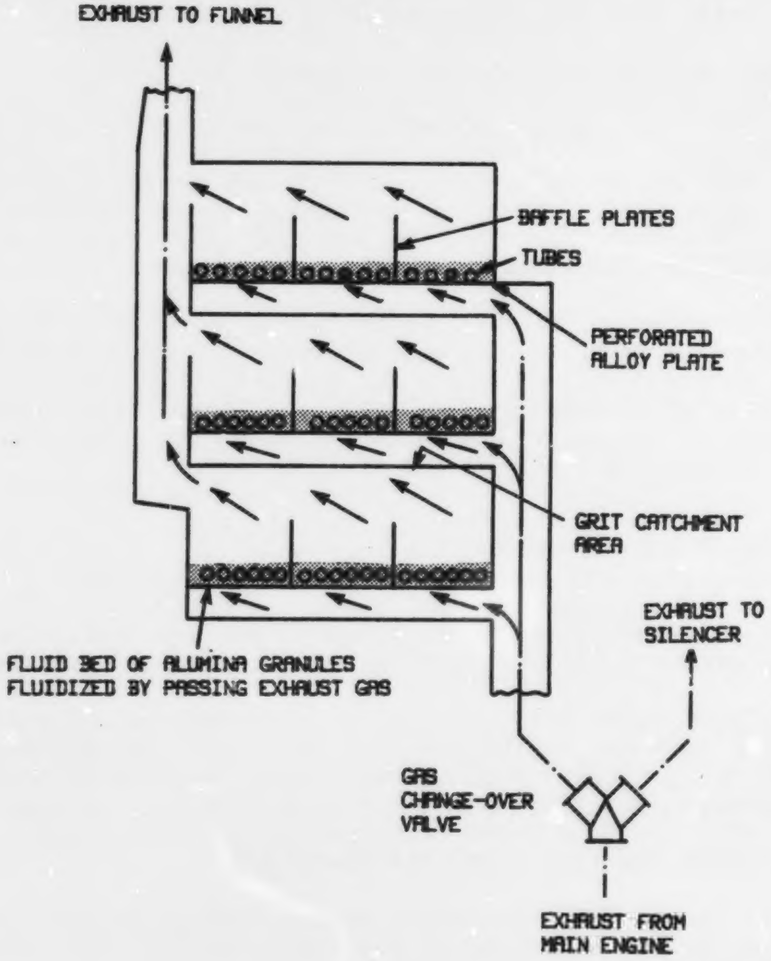
FLUIDIZED BED BOILER
(FOSTER WHEELER)


STEAM OUTLET
STEAM DRUM
GAS OUTLET
SPREADER COAL FEEDERS
BED A
BED B
MUD DRUM
LIMESTONE FEED PIPE
DOWNCOMER
FLY ASH REINJECTOR
DOWNCOMER
AIR INLET
BED MATL DRAIN (TYP)
AIR DISTRIBUTOR GRID LEVEL

[128] [134]

FIGURE 10.8

DIAGRAMMATIC ARRANGEMENT OF FLUIDIZED BED WASTE HEAT BOILER


EXHAUST TO FUNNEL
BAFFLE PLATES
TUBES
PERFORATED ALLOY PLATE
GRIT CATCHMENT AREA
EXHAUST TO SILENCER
FLUID BED OF ALUMINA GRANULES FLUIDIZED BY PASSING EXHAUST GAS
GAS CHANGE-OVER VALVE
EXHAUST FROM MAIN ENGINE

[136]

boilers the superheaters are fitted in the convection gas passage, where superheated steam temperature rapidly drops at low load. Figure 10.9 compares this characteristic for three types of boilers [133].

10.3 Combined Gas Turbine and Steam Turbine Machinery (COGAS)

A thermodynamically coupled combined gas and steam turbine installation in its simplest form uses the exhaust from a gas turbine to raise steam to drive a steam turbine. Both steam and gas turbines are coupled to provide useful power. Such plants, burning gas, light and heavy fuel oil, have been used for power generation for a number of years ashore.

The efficiency of this system is higher than that of either of the individual components, as is shown in Figure 10.10 [114]. For example, the thermal efficiency of a combined cycle land plant developed recently by Mitsubishi Heavy Industries can reach 48%. The plant called Mitsubishi Package Combined Plant (MPCP), consists of three high-temperature gas turbines exhausting into steam generators, which supply a common steam turbine. The three gas turbines and the steam turbine drive separate alternators, each connected to separate transformers [139].

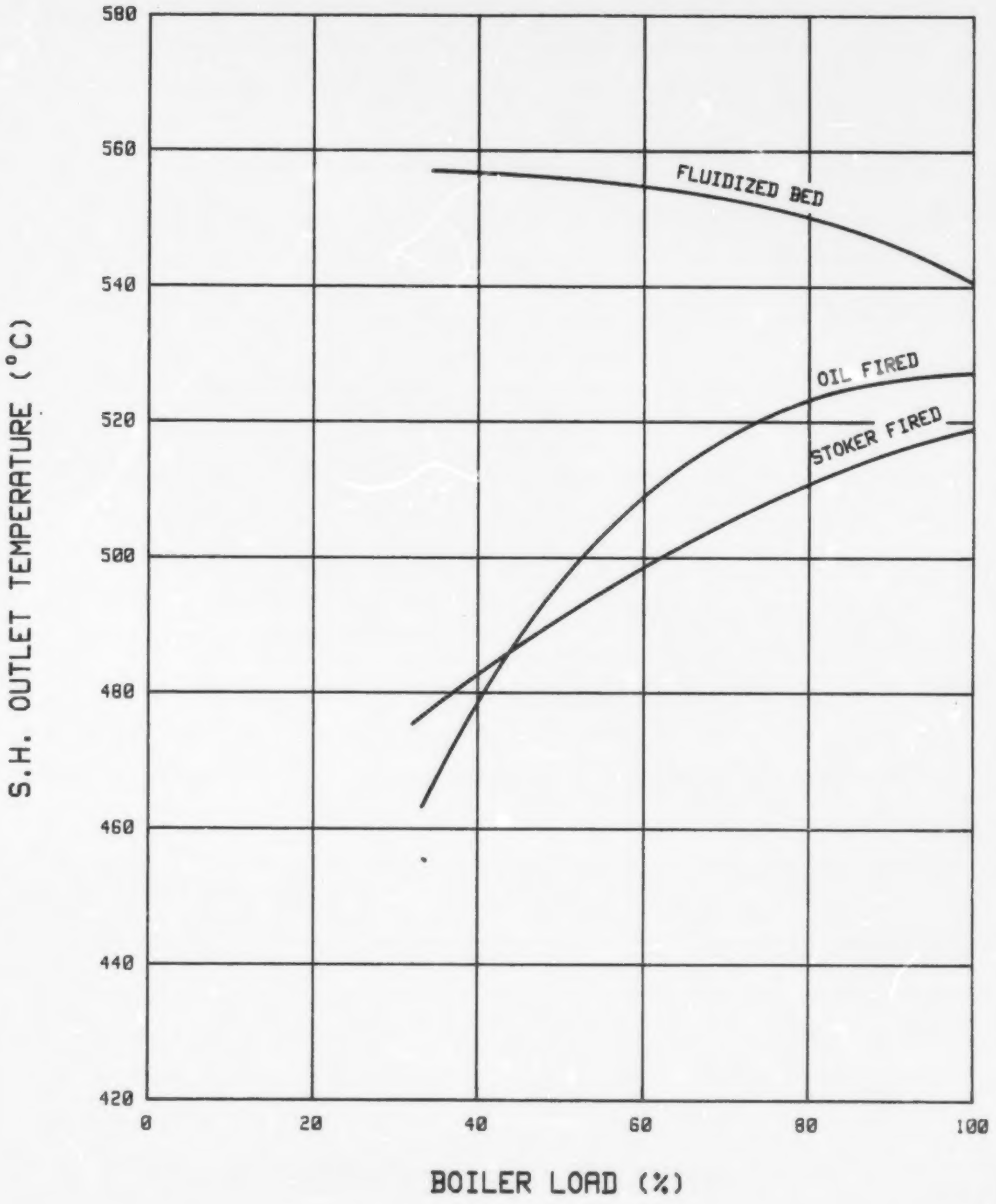
The main disadvantage of this system is the trade off between fuel quality and overhaul interval of the gas turbine. The gas turbine operates most reliably with a clean fuel supply and the outage time for overhaul increases dramatically with worsening fuel quality. However, the COGAS system may find some application for specialized vessels where clean fuel is available at a low price, e.g., boil-off gas on a LNG carrier.

Another COGAS system for marine application, without this disadvantage, was proposed by Y-ARD and GEC Gas Turbines recently. It has a heavy duty gas turbine operating with clean air as a working medium and coal as fuel and uses part of its exhaust to raise steam for the steam turbine. The other part of the exhaust is fed to the fluidized bed combustor. The cycle, which has been patented by GEC Gas Turbines Ltd., and operates as follows:

> Air Cycle: Fresh air is drawn into the compressor and delivered to a heat exchanger in the combustor, then into heat exchanger tubes laid within the fluidized bed, into the turbine driving the compressor, and finally into the power turbine. The power turbine exhausts partly into the fluidized bed and partly into the steam generator, and lastly into the atmosphere.
>
> Steam Cycle: The feed water passes through a heat exchanger, heated by the flue gases of the fluidized bed, then into a heat exchanger heated by the exhaust of the power turbine and finally to the boiler. The steam from the boiler passes to the steam turbine, exhausts into the condenser from which the condensate is pumped to the first heat exchanger.
>
> Flue Gas Path: The flue gas originating in the exhaust of the power turbine passes through the fluidized bed. After passing the flue gas heat exchanger, it is led to the funnel through a cyclone dust separator.

FIGURE 10.9

PARTIAL LOAD CHARACTERISTICS OF SUPERHEATER OUTLET STEAM TEMPERATURE *

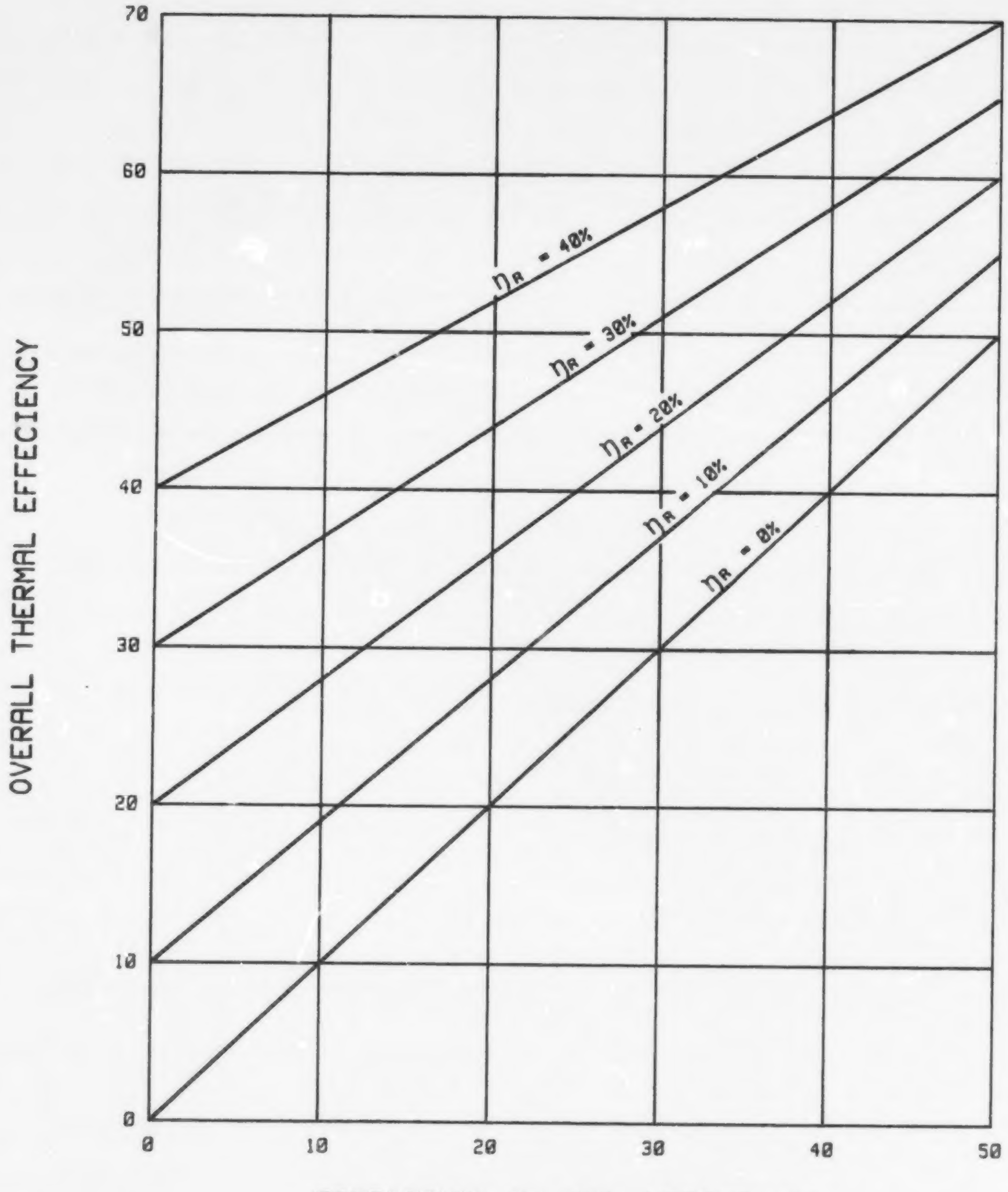


[133]

* FOR THE CASE OF NO STEAM TEMPERATURE CONTROL

FIGURE 10.10

COMBINED CYCLE EFFICIENCY



[114]

η_R = EFFICIENCY OF STEAM RANKINE CYCLE × BOILER EFFECTIVENESS

The efficiency claimed for the cycle is 34% but an upgrading of the gas and steam conditions is reputed to increase the efficiency to 37%. Both the steam turbine and the gas turbine are connected to the same shaft, geared to the propeller shaft. The plant is controlled through the gas turbine power element linked to the ship's speed setting as is the propeller pitch. The compressor speed is automatically adjusted to produce the required power level with the steam turbine throttle open. Gas turbine control is effected by bypassing the air from the compressor into the turbine inlet, hence altering the power input to the compressor turbine and changing the compressor speed, mass flow and pressure rise.

Calculations made by Y-ARD in 1980 for a bulk carrier of 50,000 dwt powered by a main engine of 9,000 kW has revealed that installation of COGAS machinery utilizing coal would be approximately 67% more expensive than slow speed diesel engine machinery and 90% more expensive than medium speed diesel machinery. However, the higher initial cost of the COGAS machinery can be overcome by savings from the differences in fuel prices. It is clear from the above that low price coal is essential if such machinery is to compete on economic terms with diesel machinery. It is likely that COGAS machinery will be better placed to compete with diesel at larger powers. It can be expected that coal burning COGAS plants can become an attractive propulsion system in the future.

10.4 Natural Gas Powered Systems

The application of natural gas for ship propulsion machinery is prevalent in LNG carriers. To date, these ships are powered by steam turbines which use steam generated in boilers burning cargo boil-off gas. Usually, the boil-off for 24 hours is less than 0.25% of the cargo.

The application of the dual fuel diesel engine in combination with lower boil-off and/or reliquefaction of the excess boil-off is claimed today as the most efficient propulsion system for LNG carriers. A machinery system proposed by Moss Rosenberg Verft and the Technical University of Norway is based on such an arrangement [140]. In this system the normal boil-off which has been assumed as equal to 0.11% per 24 hours corresponds to approximately 60% of the total fuel for the dual fuel main diesel engine in loaded conditions.

A four stage compressor with cooling after the third and fourth stage compresses the gas to a suitable pressure of approximately 200 bar and through the manifold discharges the gas to the engine. At partial load, the excess boil-off gas is delivered to the buffer tanks and additionally is stored in the spherical cargo tanks by pressure build-up to the value limited by the safety valves. The design expects that the combination of the buffer tanks and pressure increase in the cargo tanks can provide storage capacity to take care of all boil-off for a period of approximately 30 hours while the main engine is stopped. This period can be increased by using boil-off gas as a fuel for the auxiliary machinery. Installing a reliquefaction plant onboard would give greater flexibility, but it is not necessary in this design.

The diesel engine with gas injection offers the same flexibility as a steam turbine plant but the thermal efficiency is better; especially the engine with an electronic control system which can be programmed for optimum operation over the entire range of speed and the load can be very advantageous.

Figure 10.11 gives the economic comparison between various machinery arrangements of LNG carriers.

Compressed natural gas (CNG) can be used as a fuel for the engines of ships which trade on short, dedicated sea routes. A bulk carrier M.V. ACCOLADE II, which is fuelled by CNG, has been recently built in Australia and is the first ship of this type in the world. This vessel is propelled by two Fuji dual fuel diesel engines each rated at 1,230 kW, 600 r.p.m. The vessel runs daily across St. Vincent Gulf and carries limestone to a cement work at Port Adelaide. At night, natural gas, which is also the main fuel of the cement works, is compressed and loaded on the vessel. Natural gas, after being compressed to 160 bar, is charged to 21 forged steel cylinders 9 m high by 0.5 m diameter, standing in a compartment just aft of the collision bulkhead. All cylinders are connected and the gas pressure reduced to 4 bars before being piped to the engines.

The engine starts in the diesel fuel mode and after 30% load is reached it changes to gas mode. The engine operates on 93% natural gas with 7% diesel oil as a pilot fuel. In case of any abnormality during gas mode operation, automatic and instant change over to diesel mode can be made.

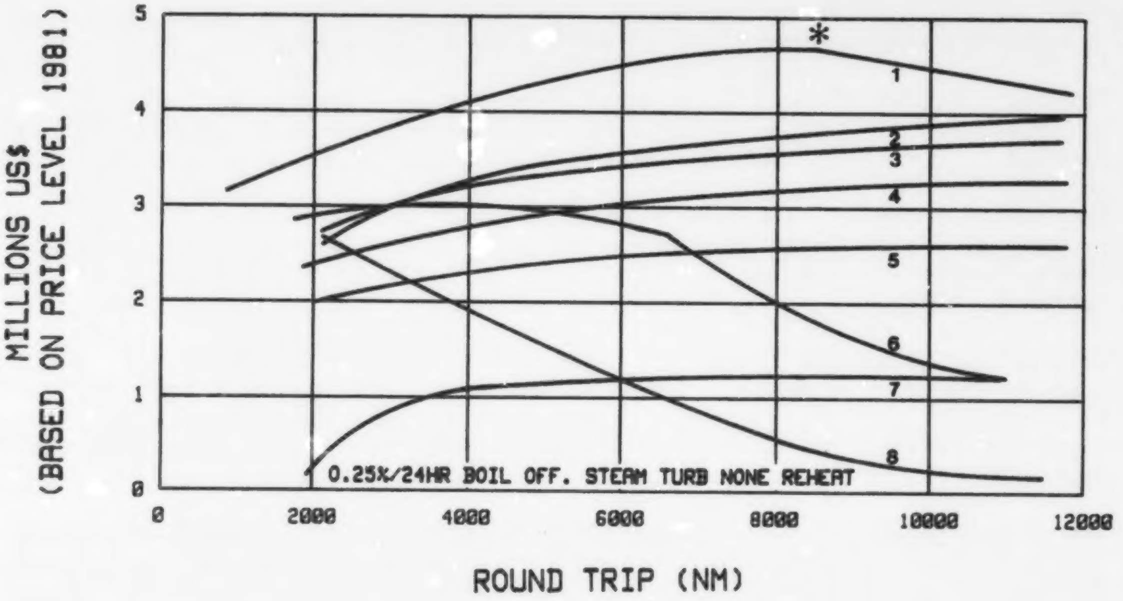
The 7,500 ton M.V. ACCOLADE II makes daily round trips of 35 nautical miles at a speed of 11 knots. A saving in the fuel cost of up to U.S. $ 250,000 per year is expected due to the large difference between the costs of natural gas and diesel fuel. The cost of the gas is based upon the bulk cost of gas to the cement plant.

As previously discussed, the high pressure gas injection diesel engine is in the course of development to overcome a drawback of conventional gas engines, namely the use of low compression ratio to overcome the problem of deterioration of the cylinder. The engine fitted to M.V. ACCOLADE II is the first of the new generation of gas engines to be fitted to a ship. Due to the larger mass of gas which is injected at high pressure into the cylinder at the end of the compression stroke, the following advantageous features are achieved:

- The engine has the highest thermal efficiency among all engines running on gaseous fuels, i.e., up to 48%.
- The engine size is reduced.
- Because of the easily controlled ignition timing, various kinds of gaseous fuels can be used.
- There is no possibility of leakage of unburned gas into the crankcase since air is alone compressed during the compression stroke.

FIGURE 10.11

NET SAVINGS PER YEAR VS. PRESENT L.N.G. CARRIERS



CURVE ASSUMPTIONS :

AVERAGE LNG DENSITY	- 0.47 ton/m^3
NOMINAL CONTROL CAPACITY	- 130,000 m^3
SPECIFIC FUEL CONSUMPTION:	
. STEAM TURBINE NONE-REHEAT	- 282 g/kWh
. STEAM TURBINE REHEAT	- 244 g/kWh
. DIESEL ENGINE	- 197 g/kWh
POWER AT 18 knots	- 24,600 kW
ENERGY COSTS:	
. HEAVY FUEL 380 cSt	- 200 US$/ton
. DIESEL OIL	- 400 US$/ton
. LNG CIF	- 280 US$/ton

CURVE No. :

1 - DIESEL ENGINE CARGO TANK PRESSURE INCREASE TO 1.25 BAR THEORETICAL

2 - 0.11%/24HR BOIL OFF. DIESEL ENGINE WITH HIGH PR. GAS INJECTION

3 - 0.11%/24HR BOIL OFF. DIESEL ENGINE PLUS RELIQUIFACTION

4 - 0.11%/24HR BOIL OFF.COMBINED CYCLE GAS TURB. STEAM TURB.

5 - 0.11%/24HR BOIL OFF.STEAM TURB. REHEAT

6 - 0.11%/24HR BOIL OFF. DIESEL ENGINE WITH CARGO TANK PRESSURE INCREASE TO 1.7 BAR

7 - 0.25%/24HR BOIL OFF. STEAM TURB. REHEAT

8 - 0.11%/24HR BOIL OFF. DIESEL ENGINE CARGO TANK PRESSURE INCREASE TO 1.25 BAR, ACTUAL

* THEORETICAL POINT WHERE SAFETY VALVE STARTS TO BLOW

In the gas engine design proposed by Mitsui Engineering fuel gas is introduced to a compressor from the gas source and after increasing pressure to 245 bar, it is supplied to a gas injection valve through a buffer tank. The power required to compress the fuel gas is approximately 0.06 kW/1 kW of the engine shaft power when the supply pressure of the fuel gas is at atmospheric pressure. In addition to the gas fuel, the liquid fuel is used by the engine by the same system as that of the usual diesel engine to be used as a pilot fuel. Moreover, the fuel oil is also supplied to an actuator pump as working fluid of the gas injection valve.

The way in which the gas is injected into the cylinder has an impact on the efficiency of the engine. Basically, two methods of gas injection can be applied. The first method, called PFI (pilot fuel injection method), employs the pilot fuel injection valve fitted in the position around the combustion chamber. The asymetrical pilot fuel jet causes unstable ignition of the main fuel while the amount of pilot fuel is reduced to a certain limit, but this limit can be kept fairly small since the hot region to be formed is limited.

In the second method called MFI (mixed fuel injection method), the pilot fuel is injected in advance into the fuel gas, and the pilot fuel and the main gaseous fuel together are injected in a mixed form through the fuel injection valve. The combustion in each cycle in this system is more even than with PFI but the minimum amount of the pilot fuel must be higher.

The investigation of heat release pattern, and hence the thermal efficiency during combustion using these two methods, have proved that an engine with PFI method is as highly efficient as a conventional diesel, while the engine efficiency using MFI is somewhat lower than marine diesel engine [121].

The operating mode of a dual fuel engine is shown by Figure 10.12. Diesel oil is used at start and stop and the engine is changed to dual fuel firing during normal operation once a minimum load is attained. When sufficient gas fuel is supplied, the ratio of the gas fuel and pilot oil may be kept at about 95:5 for stable conditions at full load. When the gas fuel supply is insufficient, the operating method is available changing the ratio of both fuels [121].

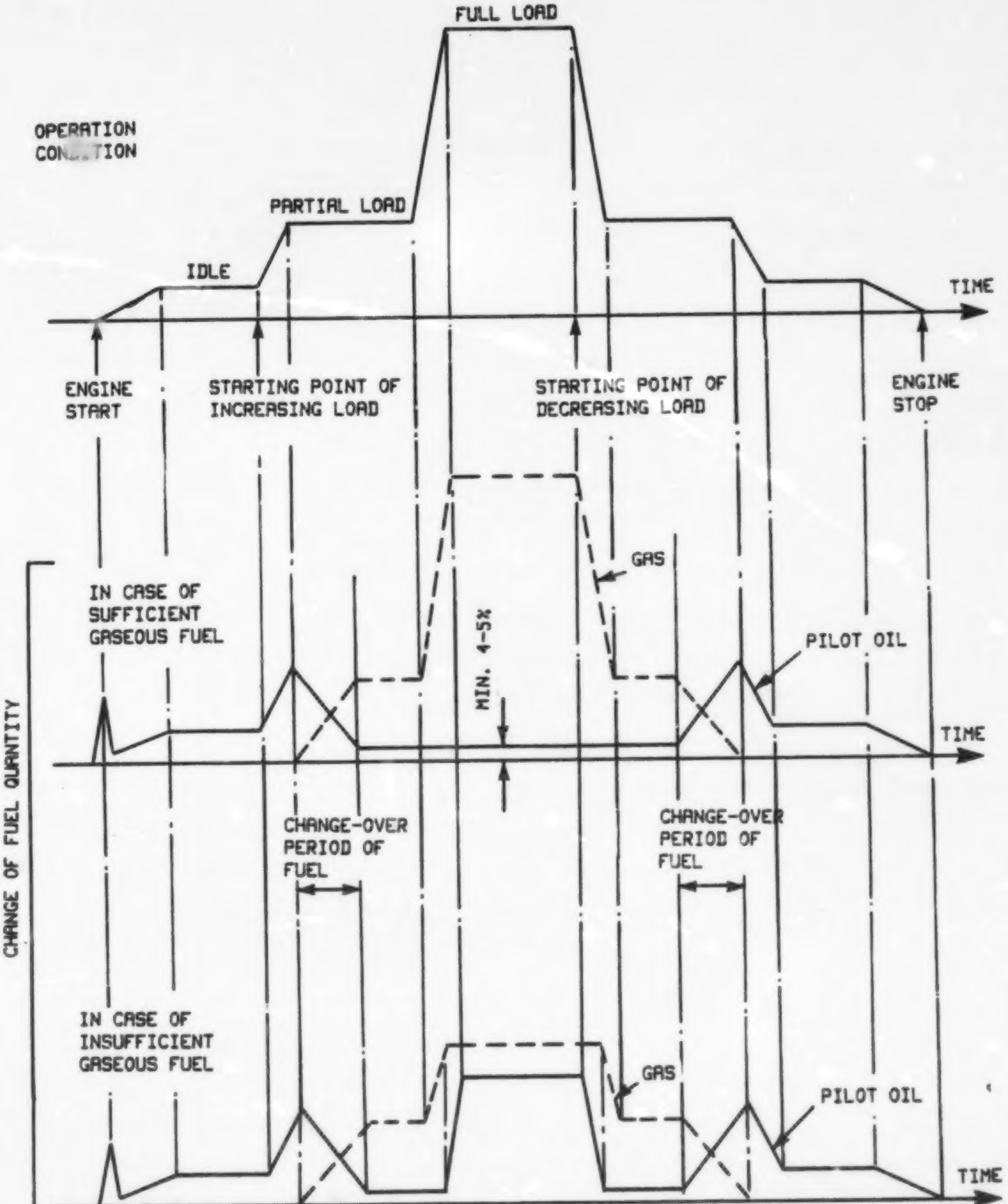
10.5 Nuclear Systems

The existing nuclear ships are almost entirely powered by the pressurized water reactors (PWR) which supply steam to steam turbines. This type of nuclear plant has the best prospects for future applications in the marine merchant shipping field. The latest arrangement, termed a Consolidated Nuclear Steam Generator (CNSG), has been developed by Babcock and Wilcox under the sponsorship of the U.S. Maritime Administration. The reactor vessel is designed as a pressure vessel to withstand the design steam pressure of the primary loop, presuming a failure in either the primary piping or the pressurized water-to-steam heat exchanger. The outline of such a system is shown in Figure 10.13 [34].

Another type of nuclear reactor which is suitable for development for use at sea is the compact gas cooled reactor (CGCR). There is a large background

FIGURE 10.12

DUAL FUEL ENGINE OPERATION MODE

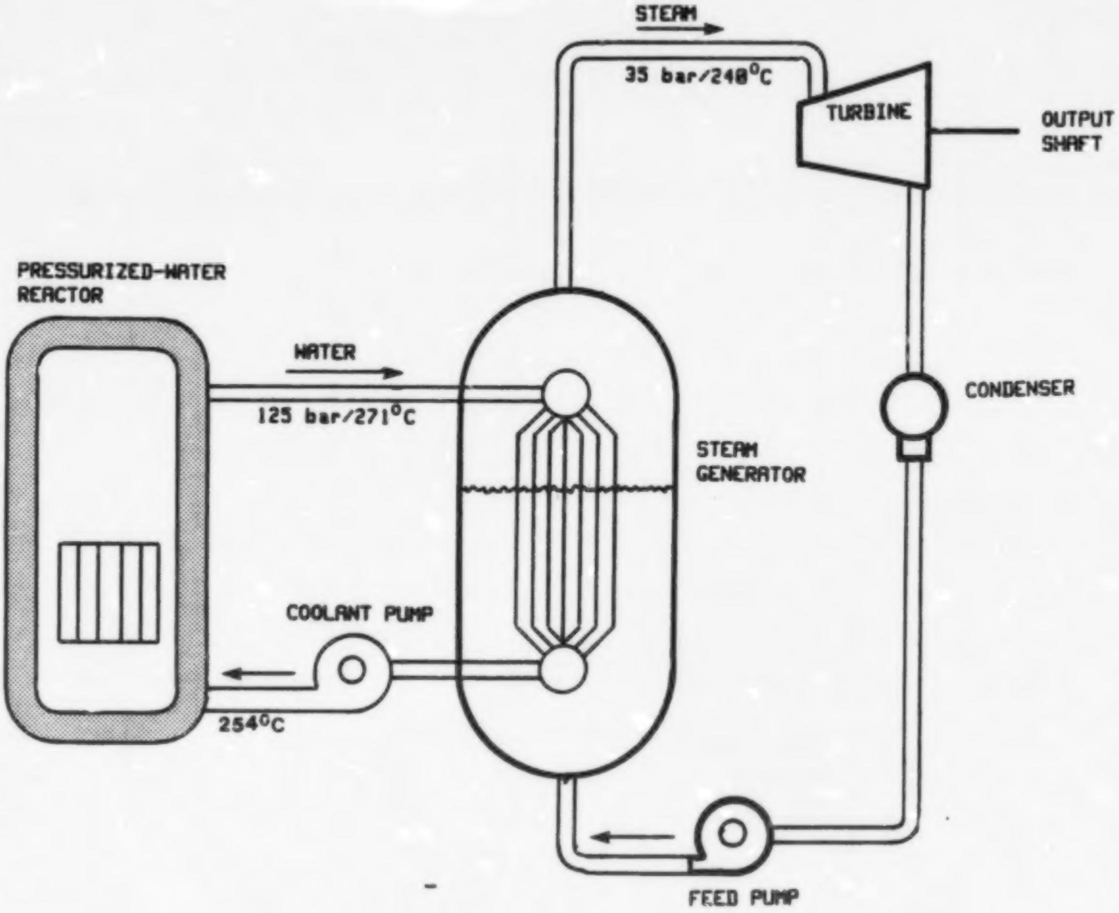

FULL LOAD
OPERATION CONDITION
PARTIAL LOAD
IDLE
TIME
ENGINE START
STARTING POINT OF INCREASING LOAD
STARTING POINT OF DECREASING LOAD
ENGINE STOP
GAS
IN CASE OF SUFFICIENT GASEOUS FUEL
MIN. 4-5%
PILOT OIL
TIME
CHANGE OF FUEL QUANTITY
CHANGE-OVER PERIOD OF FUEL
CHANGE-OVER PERIOD OF FUEL
IN CASE OF INSUFFICIENT GASEOUS FUEL
GAS
PILOT OIL
TIME
LIMITATION OF GASEOUS FUEL

[121]

MELVILLE SHIPPING LTD

FIGURE 10.13

PRESSURIZED WATER REACTOR
(PWR)


STEAM
35 bar/240°C
TURBINE
OUTPUT SHAFT
PRESSURIZED-WATER REACTOR
WATER
125 bar/271°C
CONDENSER
STEAM GENERATOR
COOLANT PUMP
254°C
FEED PUMP

[34]

of experience and technology in gas cooled reactors for land power stations as well as for military and space research applications.

Gas cooled reactors may be developed with an indirect cycle using a heat exchanger to transfer heat from the primary cooling gas to the fluid of the power generation cycle, as shown in Figure 10.14. The General Electric designs 630A, Mark IV and V, use helium for the reactor coolant and steam for the power cycle. The other possible variants of this system may use a closed cycle gas turbine instead of steam turbine. Carbon dioxide can be employed as the reactor cooling gas, as well nitrogen and oxygen. These gases have good thermodynamic properties, and may be used as the power generating fluids. Their most significant advantage is, however, that they do not become radioactive, which enables reactors with direct gas cycles to be built.

The light weight nuclear plant (LWNP), developed by Westinghouse, operates on a direct cycle using helium as the reactor cooling medium and the working fluid in a closed cycle gas turbine. This system is shown diagramatically in Figure 10.15 and an actual design in Figure 10.16. The advantages of the gas cooled reactor are; increased efficiency, improved reactor safety under casualty conditions and lower cost. This type of reactor may become a leading candidate for ship application after further development [141].

Other nuclear power cycles for marine application are possible. For example a direct boiling water reactor cycle, an organic fluid moderated reactor, a liquid metal cooled reactor (fitted to U.S. submarine SEAWOLF) and a steam cooled, heavy water moderated reactor. These cycles involve either increased risk or greater complication without sufficient advantages to compensate, therefore, they are unlikely to be applied to marine propulsion.

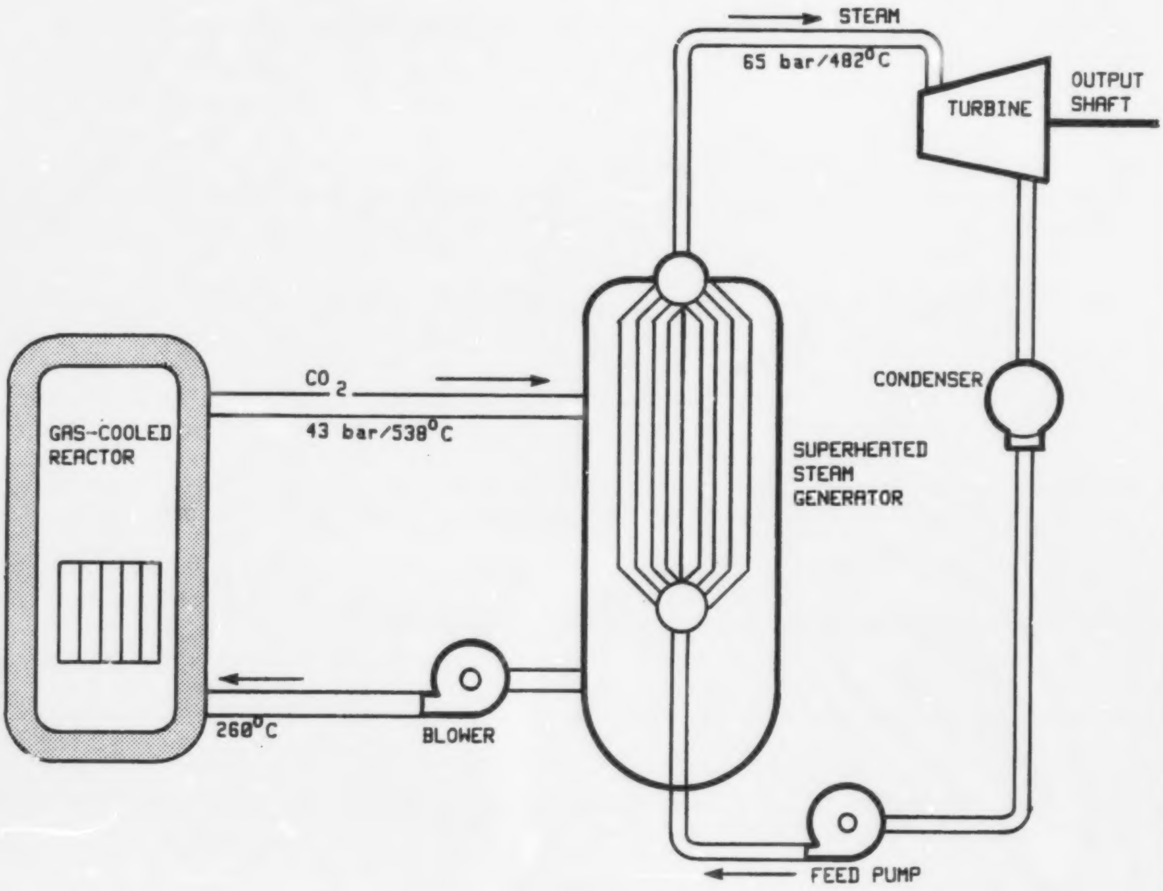
The efficiency of the nuclear power plant depends basically on the efficiency of its components, mainly the heat exchanger within the reactor and the turbine machinery. The pressurized water and boiling water reactor plants are at a disadvantage because the saturated steam conditions at the reactor steam outlet limits the steam cycle efficiency that can be attained. The gas cooled reactors are not limited by the thermal capacity of water and can operate at cycle conditions limited only by the materials of the reactor and high pressure turbine. The overall efficiency of these reactor plants are shown by Figure 10.17. For the purpose of giving an overview, Table 10.2 summarizes the basic characteristics of nuclear power plants [34].

A combination of the public's perception of environmental pollution by nuclear reactors and their high capital cost does not create any prospects for merchant nuclear ships and newbuildings in western countries in the near future. Such impediments, however, are not so significant in the Soviet Union, and the development of their nuclear merchant fleet is continuing. The fourth nuclear icebreaker ROSSIYA is presently being built in the Leningrad shipyard and it is proposed that after the completion of the icebreaker, the construction of a nuclear icebreaking lighter carrier (LASH) will be begun.

The scarcity of merchant nuclear ships and their initial stage of the practical development do not permit costs to be properly assessed. However,

FIGURE 10.14

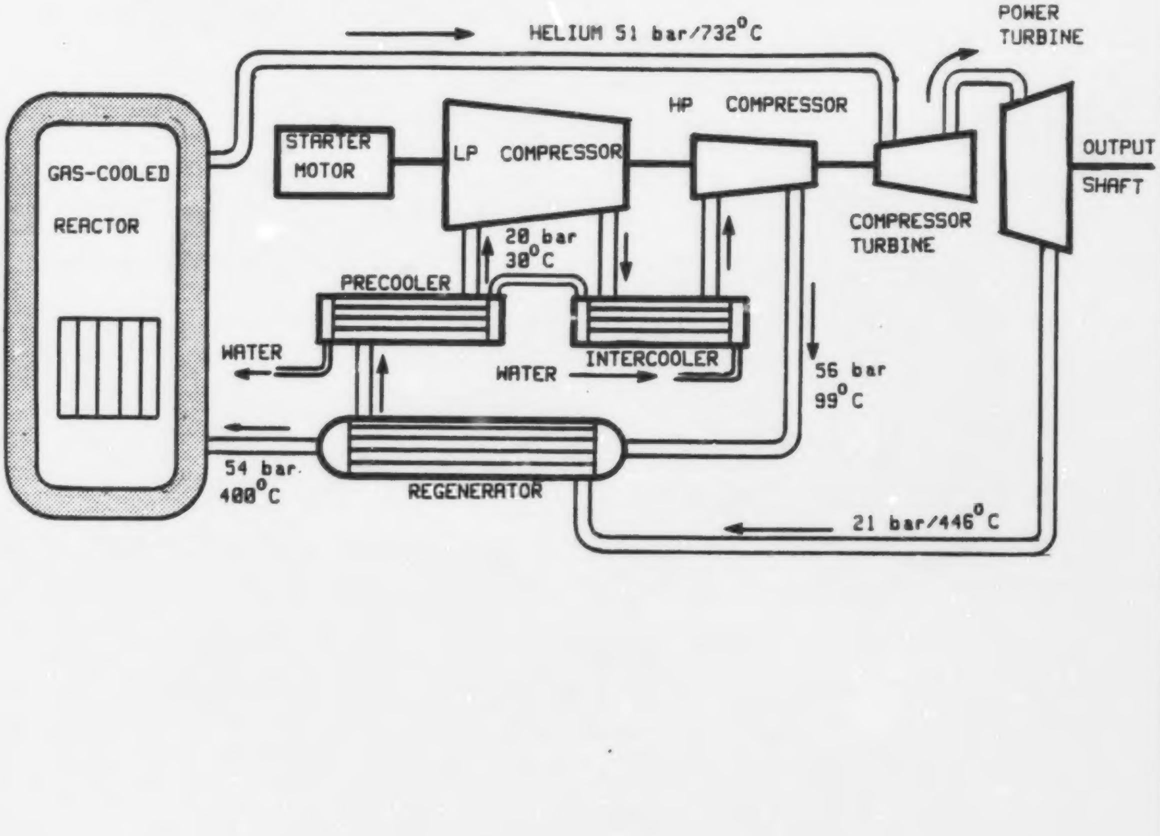
INDIRECT OR DOUBLE LOOP GAS-COOLED REACTOR WITH STEAM CLOSED CYCLE


STEAM
65 bar/482°C
TURBINE
OUTPUT SHAFT
CONDENSER
CO_2
43 bar/538°C
GAS-COOLED REACTOR
SUPERHEATED STEAM GENERATOR
260°C
BLOWER
FEED PUMP

[34]

FIGURE 10.15

DIRECT OR SINGLE LOOP SYSTEM OF GAS-COOLED REACTOR WITH GAS-COOLED CYCLE

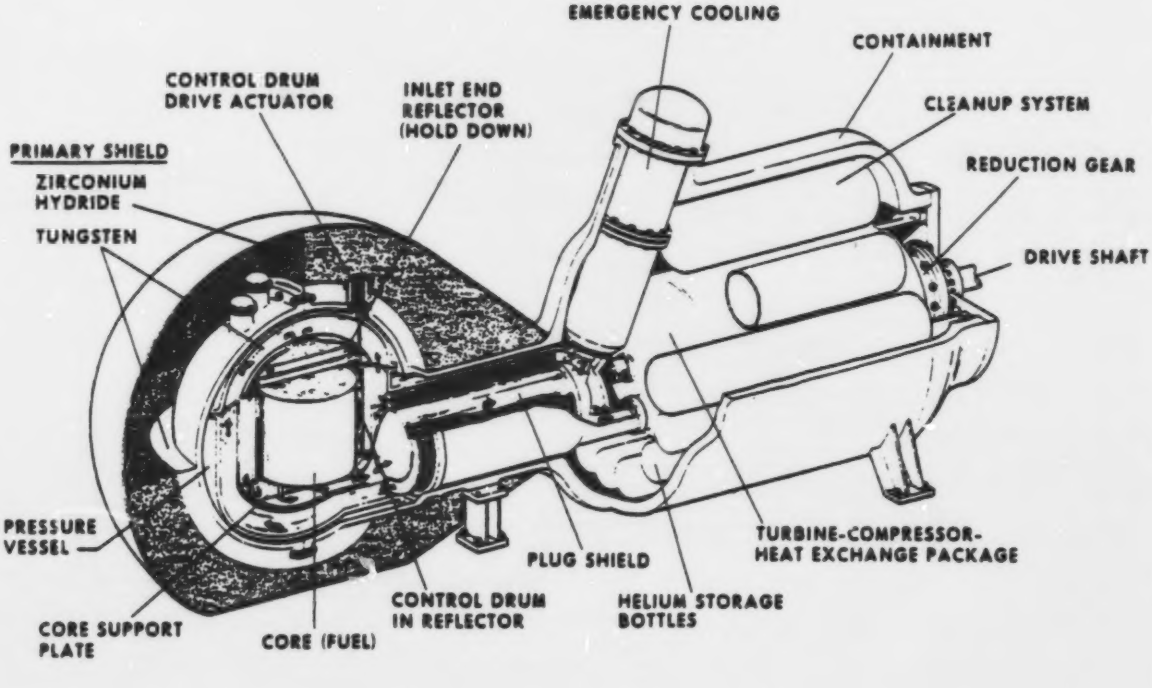


HELIUM 51 bar/732°C
POWER TURBINE
HP COMPRESSOR
STARTER MOTOR
LP COMPRESSOR
OUTPUT SHAFT
GAS-COOLED REACTOR
COMPRESSOR TURBINE
20 bar 30°C
PRECOOLER
WATER
WATER
INTERCOOLER
56 bar 99°C
54 bar 400°C
REGENERATOR
21 bar/446°C

[34]

FIGURE 10.16

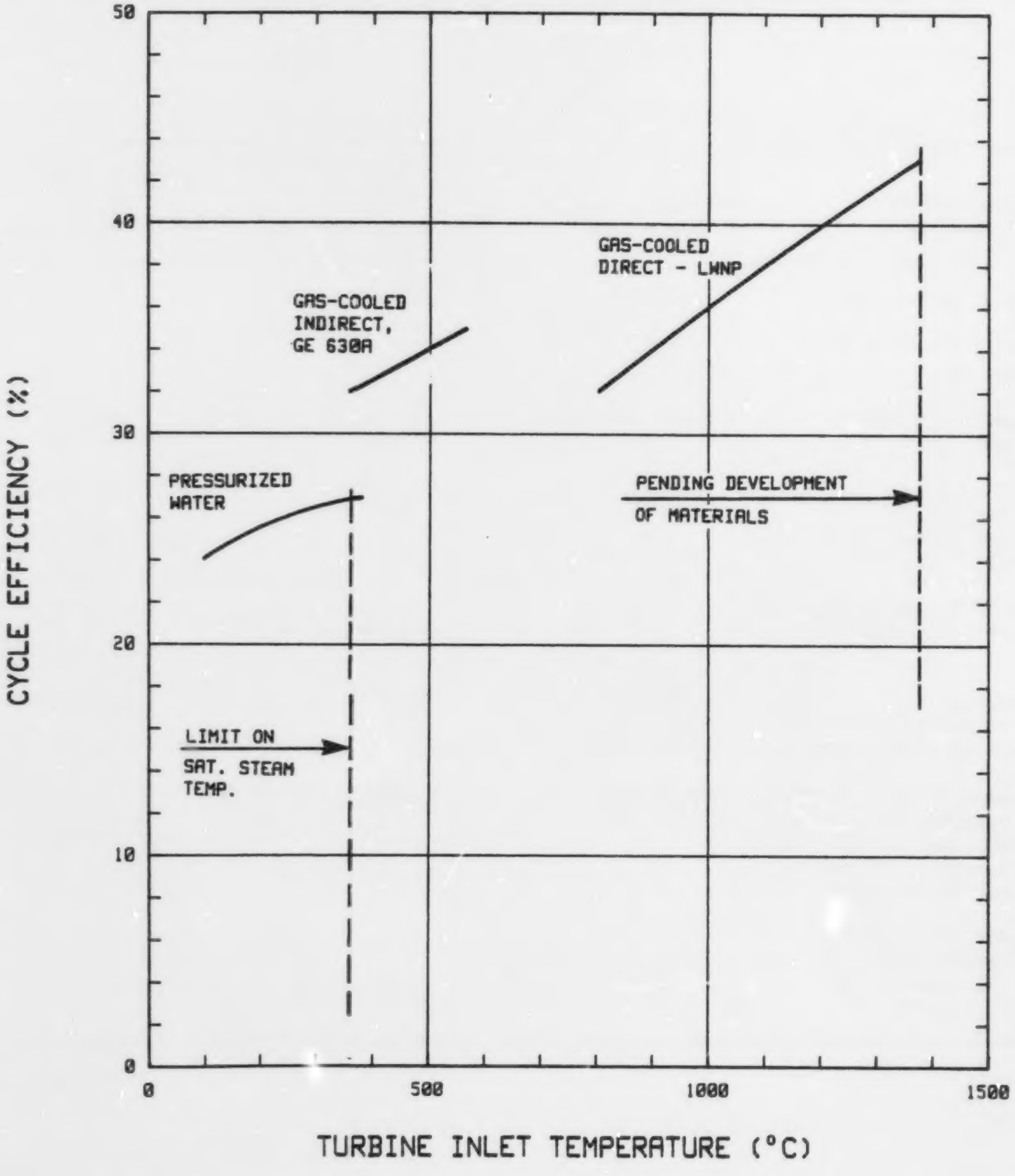
18,650 kW LIGHT WEIGHT NUCLEAR PLANT


EMERGENCY COOLING
CONTAINMENT
CONTROL DRUM DRIVE ACTUATOR
INLET END REFLECTOR (HOLD DOWN)
CLEANUP SYSTEM
PRIMARY SHIELD
ZIRCONIUM HYDRIDE
REDUCTION GEAR
TUNGSTEN
DRIVE SHAFT
PRESSURE VESSEL
PLUG SHIELD
TURBINE-COMPRESSOR-HEAT EXCHANGE PACKAGE
CONTROL DRUM IN REFLECTOR
HELIUM STORAGE BOTTLES
CORE SUPPORT PLATE
CORE (FUEL)

[141]

FIGURE 10.17

NUCLEAR POWER PLANT EFFICIENCIES



[34]

TABLE 10.2

BASIC CYCLE CONDITIONS FOR NUCLEAR REACTOR POWER PLANTS

REACTOR TYPE DESIGNATION	PRESSURIZED WATER B&W - CNSG	GAS COOLED INDIRECT G.E. 630A, MARK IV	GAS COOLED DIRECT WESTINGHOUSE LWNP
Power Rating, MW	83.5	20.4	18.6
Auxiliary Generator Load kW	6500	1446	-
Reactor Coolant	Pressurized Water	Helium	Helium
Power Cycle	Steam	Steam	Helium
Outlet Cond., bar/°C	50/280°	110/540°	107/926°
Superheat, °C	20°	225°	-
Inlet Cond., °C	204°	212°	427°
Inlet Flow, kg/hr	567×10^3	78.4×10^3	-
Reactor Thermal Load MW	314	61.3	53.6
Core Life, Years	3.5	1.98	5
Turbine Inlet, bar/°C	49/278°	108/537°	107/926°
Turbine Cylinders	2	2	2
Exhaust Cond., bar/°C	0.067/38°	0.05/33°	515°
Regenerative Feed Htg.	5 stages	5 stages	-
Compressor Inlet, °C	-	-	49°C
Compressor Outlet, °C	-	-	185°C
Compression Stages	-	-	2
Compression Ratio	-	-	3.7
Cycle Efficiency = $\frac{\text{Shaft Power}}{\text{Heat In}}$ %	26.6	33.3	34.8

[34]

some data published in the Baham Corporation study for U.S. MARAD in 1982, may give an understanding of this. Table 10.3 displays the essential cost factors for the typical nuclear power plants.

10.6 Special Fluids for Waste Heat Recovery Systems

There are two types of fluids other than water which can be employed in WHR (waste heat recovery). One type is the so called "thermal fluid" and the other type is the "organic fluid". Thermal fluids are most often used for heating purposes instead of hot water and steam. In the technical literature, thermal fluids are sometimes called organic fluids thus confusing them with the type of organic fluids used in WHR Rankine cycles.

The main advantage of thermal fluids, such as various mineral or synthetic oils, lies in their high boiling point, which permits them to carry heat at high temperatures without raising the vapour pressure. The main advantage of organic fluids for use in a Rankine cycle lies in the converse property namely in their low critical pressure and temperature. This feature allows these fluids to work within the Rankine cycle in supercritical conditions with efficiency better than in steam cycle. The reason for this improvement is explained in Figure 10.18. The supercritical cycle represents a larger fraction of the Carnot rectangle a,b,c,d than the subcritical cycle for a given heat loss q, hence its efficiency is higher. The higher efficiency of the supercritical cycle is also as the result of the higher temperature of the fluid as compared to steam cycle, this is shown in Figure 10.19 [143]. In case of the subcritical steam cycle, there is a larger "approach" temperature than with the supercritical cycle. The temperature of the single phase fluid used in the supercritical condition increases continuously throughout the boiler instead of remaining constant in the evaporation zone when latent heat is added as with a two phase fluid. The result is a smaller mean temperature difference between the exhaust gases and the working fluid, the main determinant of boiler efficiency and size, as well a higher temperature of the fluid at the outlet from the boiler.

The ideal organic fluid would have low critical conditions, low latent heat, be non-toxic, non-flammable, chemically stable and compatible with construction materials. Unfortunately, no known medium satisfies all these characteristics. Organic fluids for Rankine cycles come mainly from two groups: aromatic hydrocarbons such as toluene and chlorofluorocarbons such as freons R11, R113, R114, R21, R30, etc. The aromatic hydrocarbons are unsuitable because of flammability. R114 and R11 are the most suitable fluids. Table 10.4 gives some characteristic properties of the typical chlorofluorocarbons.

A disadvantage of the organic Rankine cycle is the poor heat transfer properties of organic fluids and a smaller mean temperature difference between the fluid and exhaust gases which results in the requirement for large and expensive boilers. Additional losses are caused by pumping a larger mass flow for the same power compared with a single steam cycle. This is because the specific heat of the fluid is about half that of steam. The organic cycle also requires a regenerator to remove superheat from the turbine exhaust to the condensate. The regenerator on the one hand improves the cycle efficiency by using otherwise rejected heat to preheat the working fluid before it enters the boiler, but on the other hand it decreases the

TABLE 10.3

REACTOR AND FUEL COST DATA

	POWER RATING (shp)		
	15000	30000	70000
REACTOR INITIAL COST FOR 1981 - U.S. $/kW			
Pressurized Water Reactor, Type CNSG			
Reactor Alone	-	-	630
Reactor and Steam Turbine Machinery	-	-	925
Gas Cooled Reactor, Indirect Cycle			
Reactor Alone	-	616	416
Gas Cooled Reactor, Direct Cycle			
Reactor and Turbo-machinery	1890	-	1005
REACTOR FUELING COSTS FOR 1981 - U.S. $/GJ			
Pressurized Water Reactor, Type CNSG	-	-	1.77
Gas Cooled Reactor, Indirect Cycle	-	2.22	2.03
Gas Cooled Reactor, Direct Cycle	2.04	-	1.95

[34]

FIGURE 10.18

IMPROVEMENT OF CYCLE EFFICIENCY
THROUGH APPLYING SUPERCRITICAL CONDITIONS

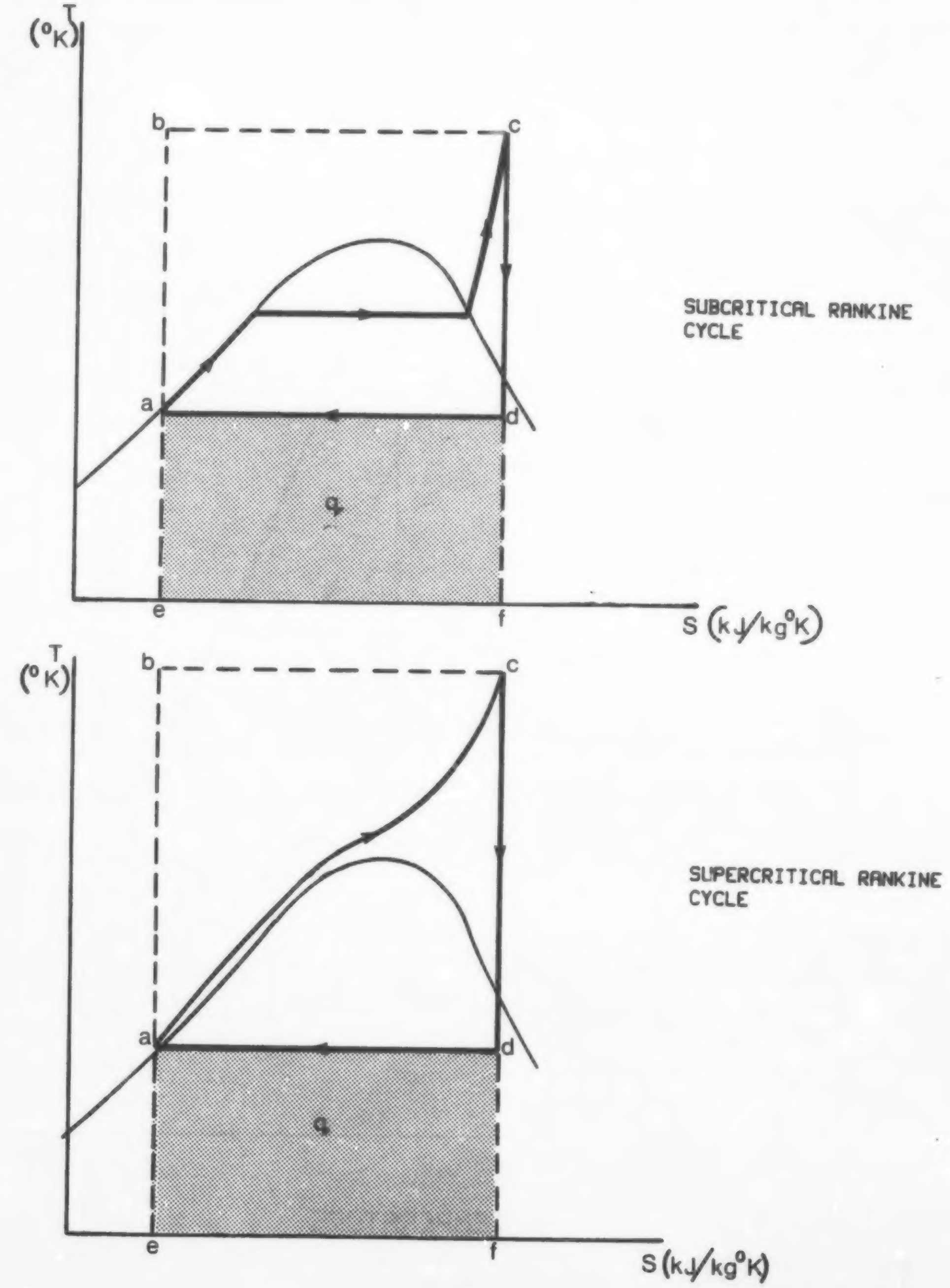
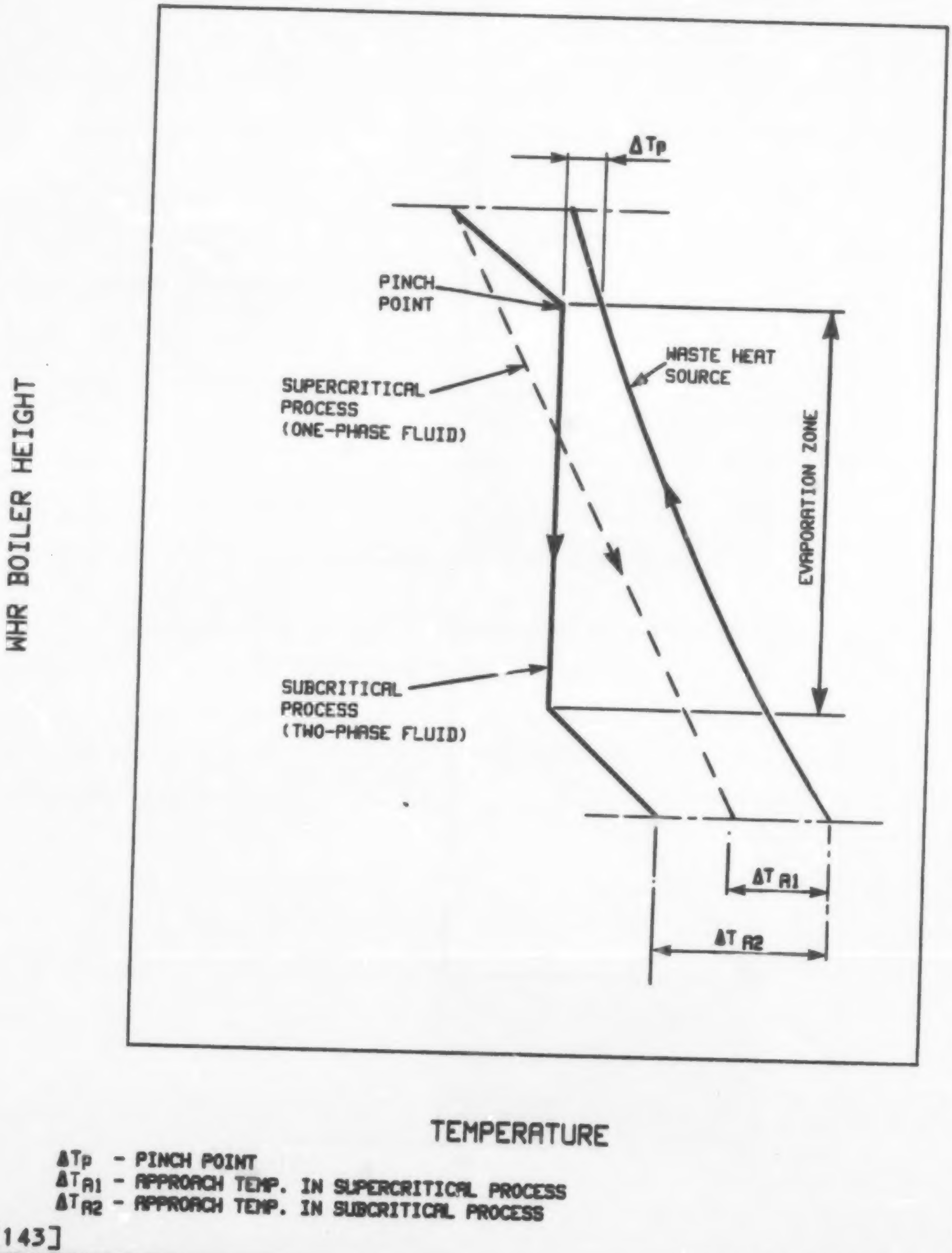

T
(°K)
b
c
a
d
q
e
f
S (kJ/kg°K)
SUBCRITICAL RANKINE CYCLE
T
(°K)
b
c
a
d
q
e
f
S (kJ/kg°K)
SUPERCRITICAL RANKINE CYCLE

FIGURE 10.19

HEAT TRANSFER CHARACTERISTICS OF SUBCRITICAL AND SUPERCRITICAL PROCESSES


ΔTp
PINCH POINT
WASTE HEAT SOURCE
SUPERCRITICAL PROCESS (ONE-PHASE FLUID)
EVAPORATION ZONE
WHR BOILER HEIGHT
SUBCRITICAL PROCESS (TWO-PHASE FLUID)
ΔT A1
ΔT A2
TEMPERATURE

ΔT_p - PINCH POINT
ΔT_{A1} - APPROACH TEMP. IN SUPERCRITICAL PROCESS
ΔT_{A2} - APPROACH TEMP. IN SUBCRITICAL PROCESS

[143]

TABLE 10.4

CHARACTERISTIC PROPERTIES OF CHLOROFLUOROCARBONS

	MEDIUM					
	R11	R12	R22	R502	R113	R114
Critical Temperature (°C)	198	112	96	82	214	146
Critical Pressure (bar)	44.0	4.12	49.8	40.8	34.1	32.6
Saturation Pressure at 30°C (bar)	1.25	7.45	11.92	13.19	0.54	2.5
Temperature at which Decomposition rate is 5% per year (°C)	150	-	-	-	200	240
Toxic or Flammable	No	No	No	No	No	No

temperature difference between the working fluid and the heating medium as well as reduces the recoverable heat from the exhaust gases. This also leads to the requirement for a larger boiler.

The organic fluids tend to break down at high temperatures and there may be restrictions imposed on the use of chlorofluorocarbons in the future due to their depletion effect on the earth's ozone layer. A ban on their production is under consideration. All these aspects are the reason why there is so little progress in the field of waste heat recovery processes using organic fluids.

There are, however, good grounds to believe that organic fluid Rankine cycles will be applied successfully in the future. They are best used for exhaust gas heat recovery. Utilizing other sources often makes organic cycles complex, relative to the benefit gained. The benefits of the use of organic fluids decrease when the source gas temperature exceeds 400°C. Above 400°C gas temperature steam provides greater efficiency.

For a system to be designed to achieve the highest efficiency from a waste heat recovery plant requires the simultaneous consideration of a large number of variables. This is best achieved by means of a computer. The Ship Research Institute of Norway has developed a program which has resulted in the system design shown in Figure 10.20. The analysis was made for MAN-B&W 10,000 kW low speed diesel engine with water cooled turbo charger. The analysis compares an organic WHR system with a conventional steam WHR system. Using a mean temperature difference across the boiler heating surfaces of approximately 30°C and the other parameters shown in Figure 10.20, an 18% increase in converted energy has been obtained by using an organic fluid instead of water. Table 10.5 shows some characteristic values which allows a comparison of the cycles [143].

A binary organic Rankine cycle (ORC) has been developed by Swedish Uddevallavarvet AB. The system is intended to utilize the low temperature part of the heat of the steam in a more efficient way than in the present WHR plants. Figure 10.21 is a schematic diagram of a binary Rankine cycle. A screw expander instead of turbine has been proposed for ORC cycle, as it is less sensitive to moist vapour which may result during part load operation. The analysis of this sytem has revealed that the binary cycle installed on the ship with B&W engine of 13,500 kW would yield about 46 kW or 11% more net output at a steam turbine back pressure of 0.5 bar than the single steam cycle. However, the extra cost of the binary cycle relative to the steam cycle would be in the range of $235,000 (Cdn.) which cannot at present be justified in financial terms. The cost of this system as well as that of the other organic Rankine cycles include development costs. This means that in case of series production economies can be gained [144].

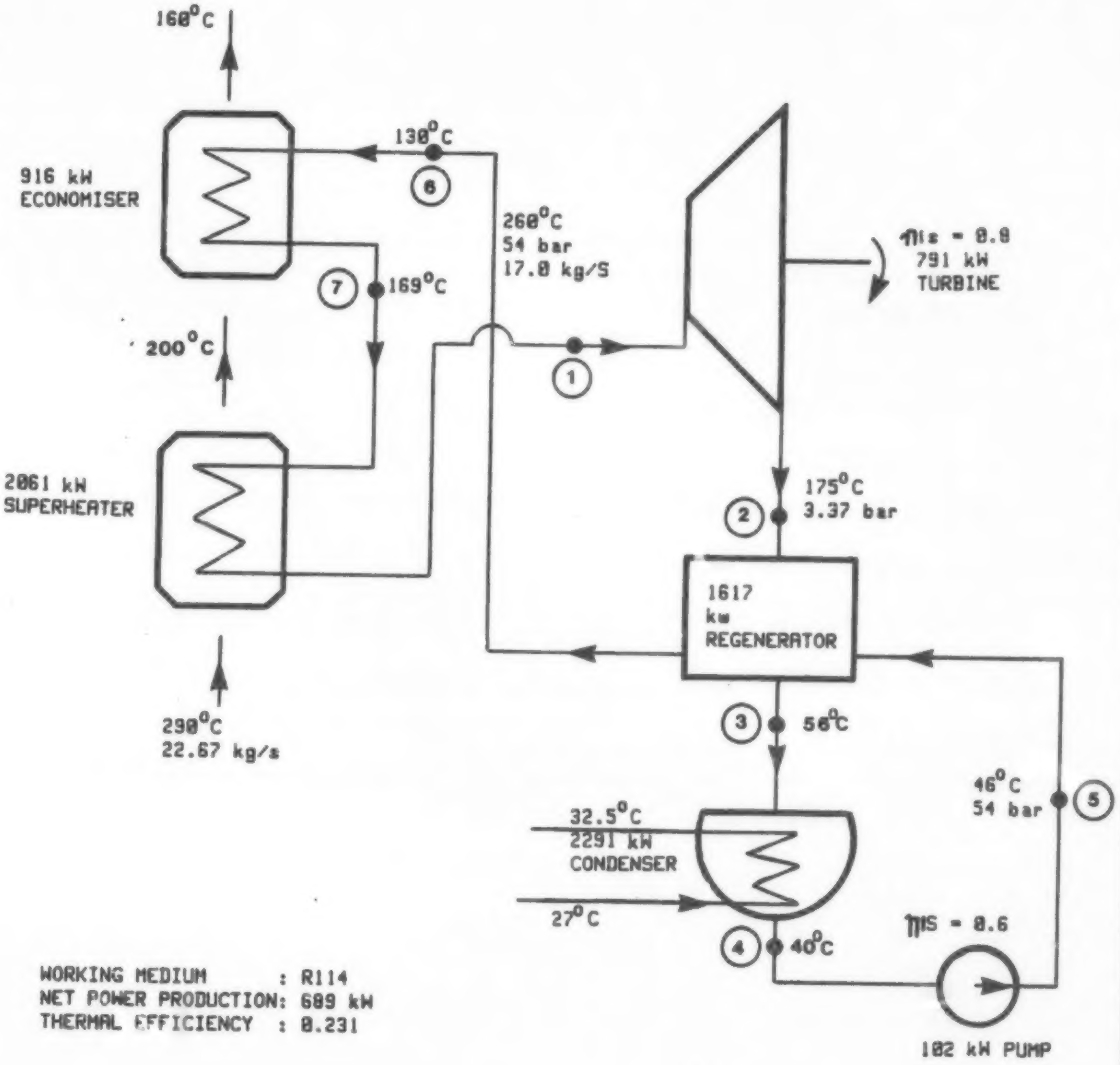
The other application for the ORC which have been studied in Japan are:

- direct contact heat exchangers (oil/freon) where oil absorbs heat in an exhaust gas boiler and is mixed with the freon, which is led to a turbine (the Hitachi OFT system) and;

- solar energy collectors as vapour generators in a cycle where the turbine drives a compressor in an air conditioning plant.

FIGURE 10.20

ORGANIC RANKINE WHR SYSTEM


160°C
130°C
⑥
916 kW
ECONOMISER
260°C
54 bar
17.0 kg/S
ηis = 0.8
791 kW
TURBINE
⑦
169°C
200°C
①
2061 kW
SUPERHEATER
175°C
3.37 bar
②
1617
kw
REGENERATOR
290°C
22.67 kg/s
③
56°C
46°C
54 bar
⑤
32.5°C
2291 kW
CONDENSER
27°C
ηIS = 0.6
④
40°C
102 kW PUMP

WORKING MEDIUM : R114
NET POWER PRODUCTION: 689 kW
THERMAL EFFICIENCY : 0.231

[143]

TABLE 10.5

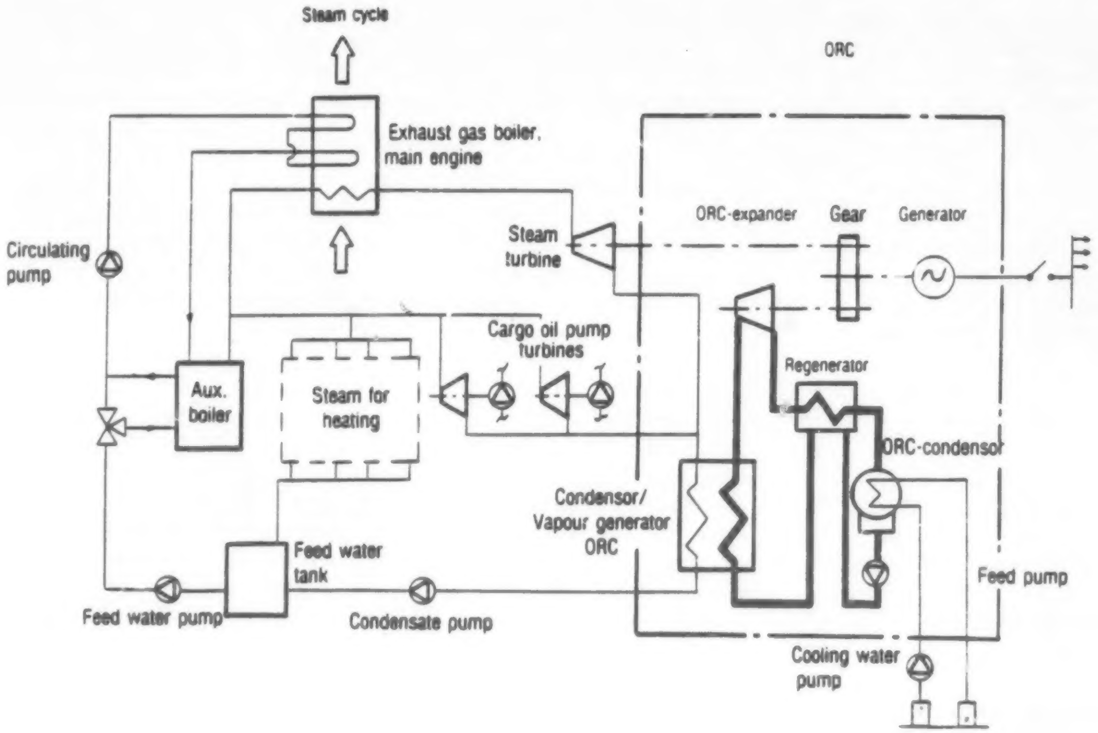
COMPARISON BETWEEN CYCLES

CYCLE PARAMETERS	STEAM CYCLE	R114 CYCLE
Superheating Temperature	260°C	260°C
Boiling Pressure	7.0 bar	54 bar
Condensing Pressure at 40°C	0.075 bar	3.372 bar
Fluid Flowrate	1.10 kg/s	17.04 kg/s
Net Power Production	584 kW	689 kW
Total Area of Boiler	2080 m^2	2570 m^2
Thermal Efficiency	0.189	0.231

[143]

FIGURE 10.21

BINARY RANKINE CYCLE


Steam cycle
ORC
Exhaust gas boiler, main engine
ORC-expander
Gear
Generator
Steam turbine
Circulating pump
Cargo oil pump turbines
Regenerator
Aux. boiler
Steam for heating
ORC-condensor
Condensor/ Vapour generator ORC
Feed water tank
Feed pump
Feed water pump
Condensate pump
Cooling water pump

[144]

These projects are more sophisticated but should be taken into account while planning energy efficient systems of the future.

10.7 Heat Pumps

Heat pumps are based on the well known refrigerator principle, where heat is taken from a low temperature source and dissipated to a higher temperature coolant. Heat pumps in some respects are analogous to hydro electric pumped storage systems. In these systems water at a lower level is pumped to a higher level and the energy used to carry out the pumping can be regained at a later stage. In a pumped storage scheme the energy input is equal to that recovered whereas the heat pump is able to recover several times its energy input. This difference is because while the working medium of the pumped storage scheme (water) remains in the same state throughout the cycle, in the heat pump, the medium (liquefied gas) changes from the liquid to the vapour phase and back during the cycle. Thus, in the process, much more energy is exchanged than is needed to drive the heat pump. This can be seen in Figure 10.22 where the theoretical cycle diagram has been shown for a compression heat pump. All considerations here are confined to the compression heat pumps, since they are far superior economically to the absorption heat pumps.

The performance of a heat pump is assessed by relating the useful energy Q_{out} to the energy input Q_{in}. With heat pump values above 3 are normal because only the compression work W weighs in the energy input, and not the heat Q_o absorbed by the evaporator, which is available without cost. This exploitation ratio is termed the coefficient of performance or performance energy ratio (ε) :

$$\varepsilon = \frac{Q_{out}}{Q_{in}} = \frac{Q_o + W}{W} = 1 + \frac{Q_o}{W}$$

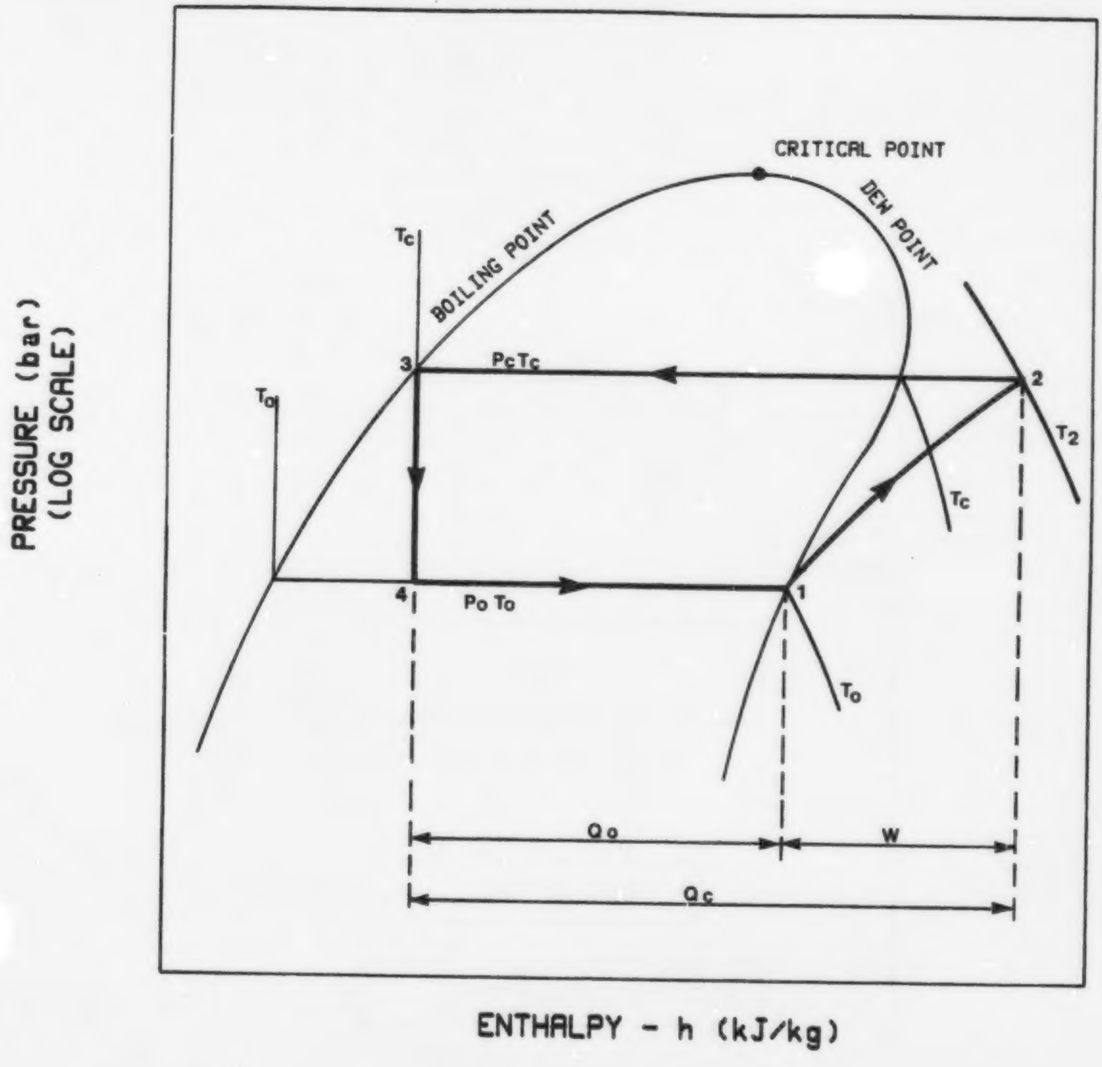
From Figure 10.22 it can be seen that by lowering T_c or increasing T_o, Q_o/W can be increased, and hence a better coefficient of performance can be obtained. Therefore, heat pumps can be used to convert low grade heat, i.e., heat at low temperature, to high grade heat,(heat at high temperatures) and thus permit the recovery of previously unuseable heat [145].

The temperature difference between the heat source and the heat user in ship recovery systems, can be in the range of 10 to 20°C, which makes the design economically justified. The application of heat pumps is therefore claimed to be profitable when used for heating purposes on small ships.

As an example, the heating system shown in Figure 10.23 will be described [146]. The diesel engine cylinder jacket cooling water has been selected as a heat source because the water temperature is more stable than that of the exhaust gas. The temperature of the cylinder jacket cooling water is in the range of 68 to 78°C and the heat pump utilizes the heat in this water to raise the temperature of the water in the domestic heating system to 80 to 90°C. Water from the heat pump could also be used for heating oil in settling and service tanks, therfore, the use of the auxiliary boiler can

FIGURE 10.22

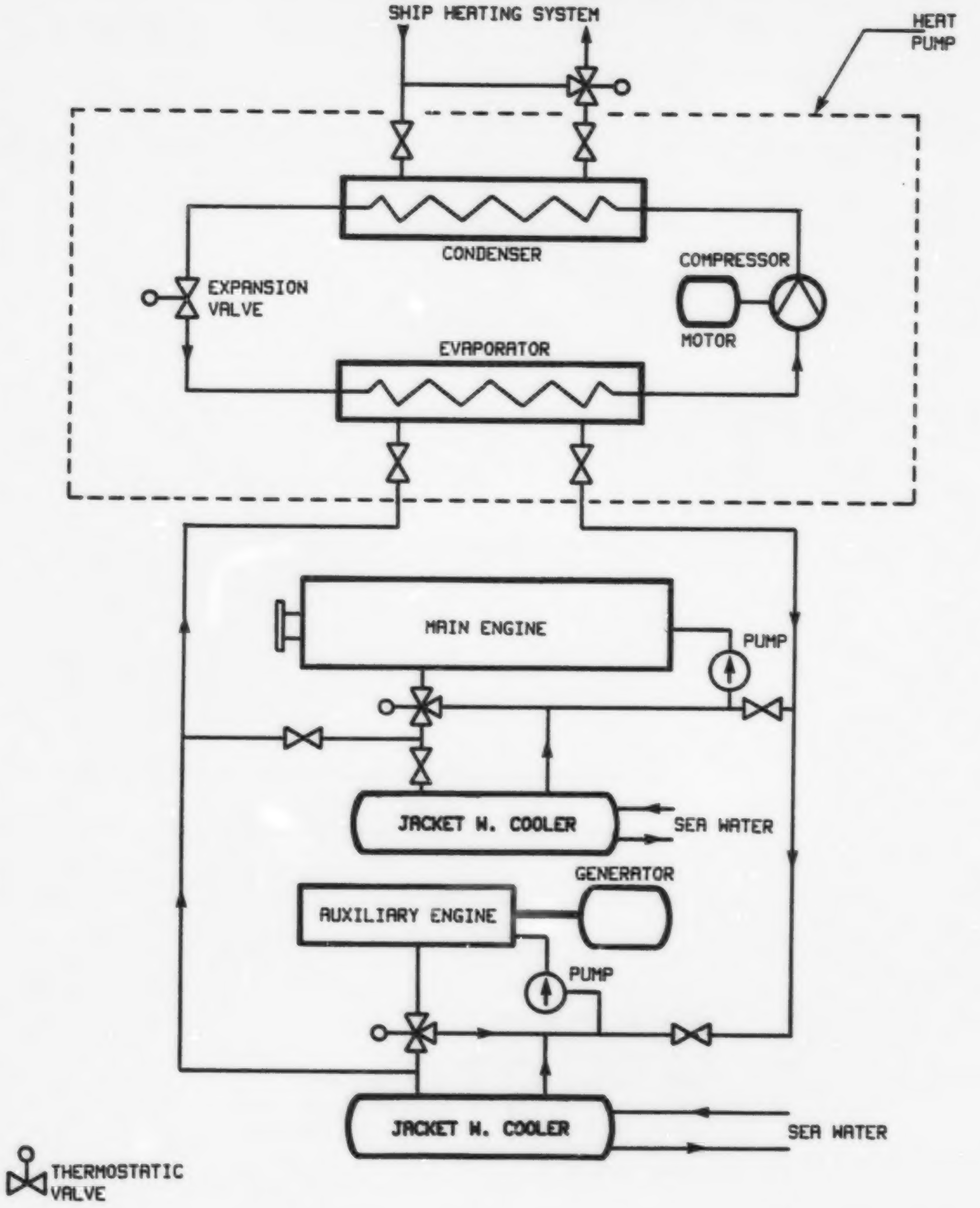
THERMODYNAMIC CYCLE OF HEAT PUMP


CRITICAL POINT
BOILING POINT
DEW POINT
Tc
To
3
PcTc
2
T2
Tc
4
Po To
1
To
Qo
W
Qc
PRESSURE (bar) (LOG SCALE)
ENTHALPY - h (kJ/kg)

P_cT_c - CONDENSER PARAMETERS
P_oT_o - EVAPORATOR PARAMETERS
Q_o - HEAT YEILDED BY HEAT SOURCE AT LOWER TEMPERATURE T_o IN THE EVAPORATOR
Q_c - HEAT TRANSFERRED IN THE CONDENSER FROM THE WORKING MEDIUM TO THE HEAT CARRIER AND THEN IN TURN TO THE CONSUMER
W - COMPRESSION WORK

FIGURE 10.23

HEAT PUMP APPLICATION
FOR SHIP HEATING PURPOSES


SHIP HEATING SYSTEM
HEAT PUMP
CONDENSER
COMPRESSOR
EXPANSION VALVE
MOTOR
EVAPORATOR
MAIN ENGINE
PUMP
JACKET W. COOLER
SEA WATER
GENERATOR
AUXILIARY ENGINE
PUMP
JACKET W. COOLER
SEA WATER
THERMOSTATIC VALVE

[146]

be eliminated at sea. By using in-line water/oil heaters, the greater part of the heat required for both purifying and burning can also be obtained and decrease the amount of electrical power required.

The heat pump equipment comprises an evaporator, electrically driven compressor, condenser and expansion valve. The working medium would be freon or freon/oil mixture which is non-corrosive, inert and non-toxic. Since there is waste heat available in the engine jacket cooling water at a temperature very close to that required to heat the oil, the power of the heat pump compressor would be low. The coefficient of performance of this system, i.e., ratio of energy out to energy expended, can easily be 7 which should accrue a significant economic benefit.

The relatively high cost of the equipment has until now discouraged its use at sea despite the acknowledged benefits. As fuel prices rise, the capital cost of the heat pump equipment can be amortized more quickly and the use of this equipment should gain in popularity.

10.8 Closed Cycle Gas Turbine

The thermodynamic cycle consisting of two isentropic processes and two constant pressure processes which uses a gas as the working fluid is known as the Joule or Brayton cycle. This cycle is applied to gas turbine plants. There are two types of gas turbine engine cycles:

- open cycle with fuel burned directly in the working fluid
- closed cycle with fuel burnt externally to the working fluid

The open cycle gas turbines are the most commonly used. In this turbine the atmospheric air is drawn into the circuit continuously and energy is added by the combustion of fuel in the working fluid. Products of the combustion are exhausted into the atmosphere. In case of closed cycle gas turbine engines, the same working fluid is repeatedly circulated through the machinery and the necessary energy is added to the fluid in a heater. The heat is obtained from fuel burnt in a separate air stream. Figure 10.24 shows the basic closed cycle. This type of gas turbine was developed before World War II but has not been extensively used in practice principally because of the abundance of high grade, clean fuels which were available and could be used in the open cycle alternative. As the closed cycle gas turbine has the advantage of being able to burn "dirty" fuels this engine may become a serious competitor of contemporary marine propulsion power plants in the future.

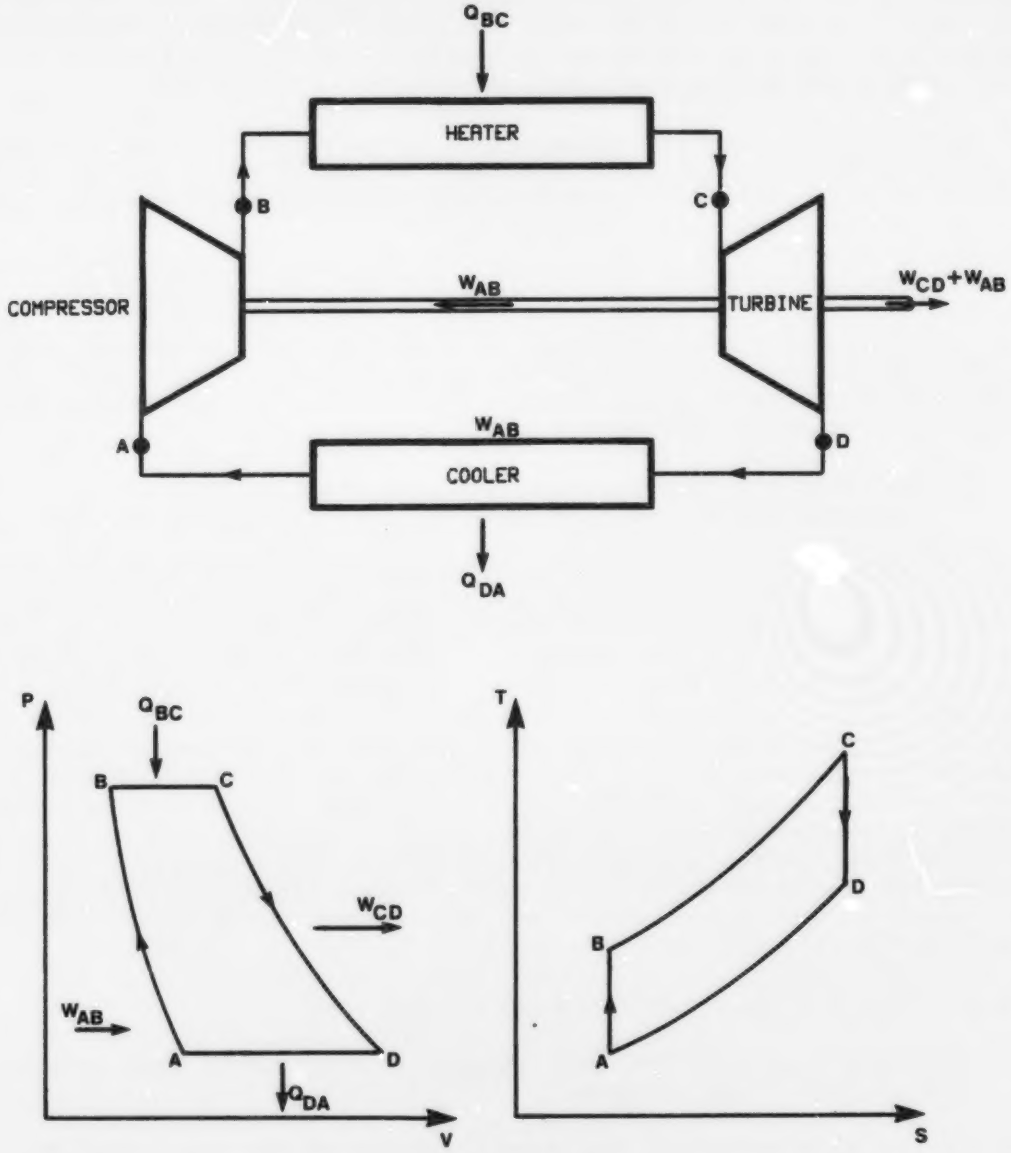
The advantages of the closed cycle gas turbine are:

- There are no limitations with respect to the type of fuel or the type of source heat exchanger.

- By the elimination of the direct contact of the fuel with the turbine, the turbine is not affected by harmful contaminants of combustion, thereby the life of the turbine will be increased and higher cycle temperatures can be permitted.

FIGURE 10.24

DIAGRAMMATIC REPRESENTATION
OF THE CLOSED BRAYTON CYCLE



Q_{BC}
HEATER
B
C
COMPRESSOR
W_{AB}
TURBINE
$W_{CD}+W_{AB}$
W_{AB}
A
D
COOLER
Q_{DA}
P
Q_{BC}
B
C
W_{CD}
W_{AB}
A
D
Q_{DA}
V
T
C
D
B
A
S

- With a pressurized system, the power output can be altered by a change of pressure level in the circuit and hence the mass flow of the fluid. This form of control means that a wide range of load can be accommodated without alteration of the maximum cycle temperature and therefore with little variation of overall efficiency.

The disadvantages of the closed cycle gas turbine are:

- The quality of heat rejected from the system will be a function of "pinch point", i.e., the difference of temperature between the exhaust gases leaving the system and the lowest temperature of the working fluid. However, normal boiler efficiencies will apply to the heat exchanger.

  The allowable working temperature of the surfaces in the heater will impose the upper limit on the maximum temperature of the main cycle. So the usual problem of high temperature corrosion remains unsolved for the combustion chamber.

- The closed Brayton cycle is more complicated compared with open cycle, but is much simpler than a steam cycle.

- The use of a working medium, other than air, imposes additional costs.

- Rotating seals of the compressor and turbine are required to work at high temperature and pressure. These parts are difficult to construct and maintain.

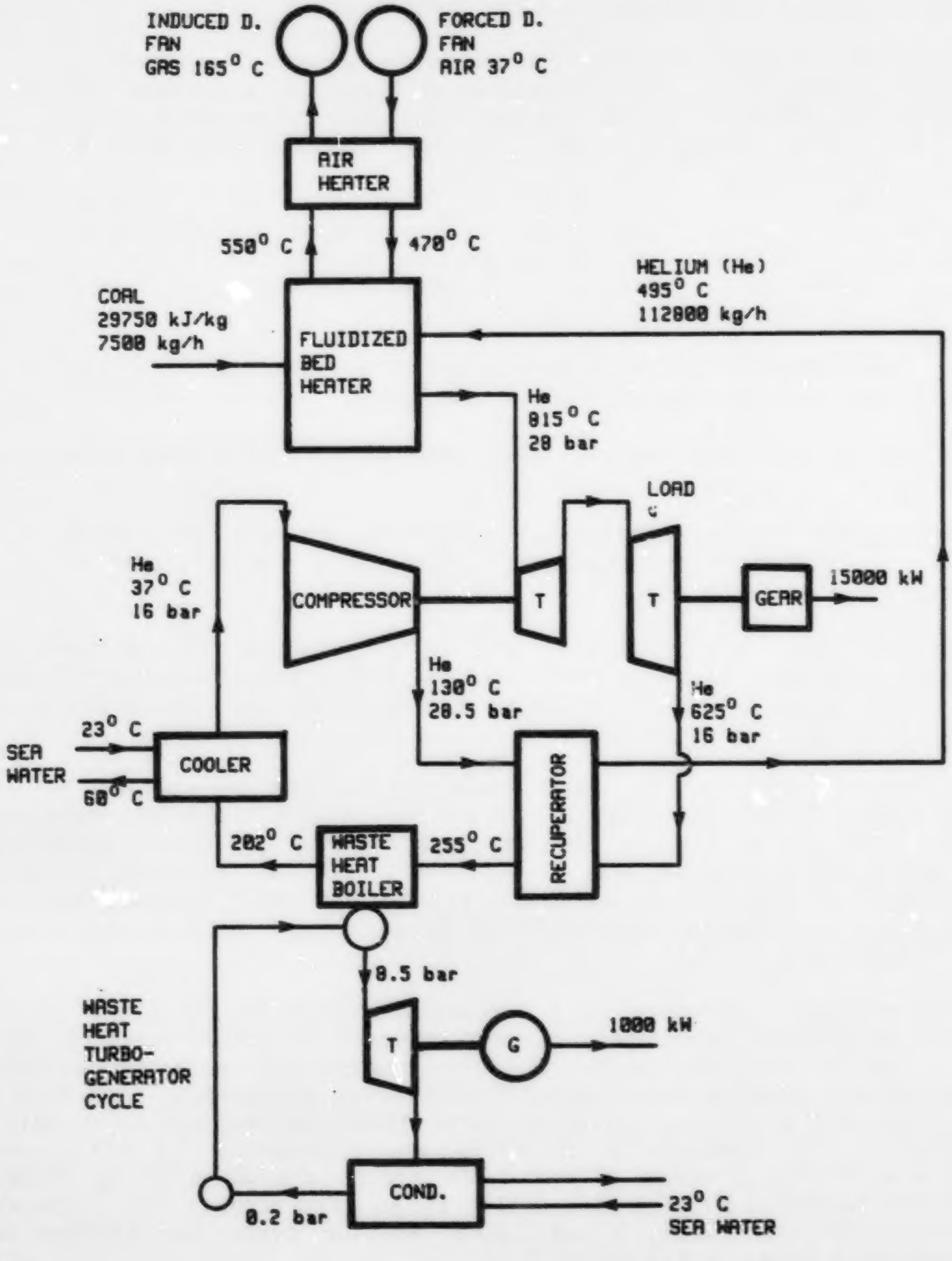
These disadvantages are of a practical nature and can be resolved by development. Interest is being revived in the closed Brayton cycle turbines as a future means for the propulsion of ships due to its ability to efficiently utilize low grade fuels.

Although research work on the closed cycle gas turbine was carried out in the 1950 - 1960 period, the cost savings did not justify the effort required for development due to the low cost of clean fuel at the time. Recently, more research and development work is believed to be underway which has been brought about by the need to efficiently burn low grade fuels. Many of these research undertakings are sponsored by government agencies in U.S.A. and other countries.

Since the principal component in a successful design of the closed cycle system is the combustion chamber, current research is concentrated on this subject. The latest proposals for oil fired systems use radiant combustors in which ceramic materials such as silicon nitride and silicon carbide are incorporated, while for coal fired systems, fluidized bed combustor which can withstand the temperature of 950°C are proposed. The efficiency attainable with the proposed closed cycle plants are as high as 47.2%, although the system is still in the design stage. One project for 15,000 kW ship propulsion plant using a coal fired Brayton cycle gas turbine is diagrammatically shown in Figure 10.25.

FIGURE 10.25

CYCLE DIAGRAM FOR A CLOSED BRAYTON CYCLE COAL-FIRED MARINE PLANT OF 15,000 kW


INDUCED D.
FAN
GAS 165° C
FORCED D.
FAN
AIR 37° C
AIR
HEATER
550° C
470° C
HELIUM (He)
495° C
112800 kg/h
COAL
29750 kJ/kg
7500 kg/h
FLUIDIZED
BED
HEATER
He
815° C
28 bar
LOAD
He
37° C
16 bar
COMPRESSOR
T
T
GEAR
15000 kW
He
130° C
28.5 bar
He
625° C
16 bar
SEA
WATER
23° C
60° C
COOLER
RECUPERATOR
282° C
WASTE
HEAT
BOILER
255° C
8.5 bar
WASTE
HEAT
TURBO-
GENERATOR
CYCLE
T
G
1000 kW
COND.
0.2 bar
23° C
SEA WATER

[34]

initial costs of gas turbine system components are well known from open cycle turbine production experience and the cost of a closed cycle plant may be accurately calculated. For example, the Baham Corp. study estimates initial cost of closed Brayton cycle turbine for a 30,000 kW merchant ship propulsion utilizing heavy fuel oil, as equal to $390 U.S. (1981) per kW [34].

10.9 Stirling Cycle Engines

The Stirling cycle engine has for some time been only of historical interest. The engine was invented in 1816 and other than a few prototypes, it has not been used for practical power producing systems. Recent increases in fuel prices and restrictions on air pollution have again concentrated interest in the engine and machines based on the Stirling cycle have been built and successfully operated. This development has been partially due to the possibility of using this engine for power production in space, however, they also have promising prospects for marine application.

These externally heated piston engines work with a closed gas cycle. The gas is alternatively heated and cooled. When heated, the gas expands and acts upon the piston; when cooled it permits the piston to return. The elements of the thermodynamic cycle are shown in Figure 10.26. The cycle consists of two isothermal and two constant volume processes. The heat supplied during process B-C is equal in quantity to the heat rejected during process D-A. Ideally, this operation is reversible and is accomplished by use of a heat regenerator. Figure 10.26 shows simplistically the principles of the system. The engine consists of a cylinder with two pistons and a regenerator R, formed from a plug of very fine wire gauze, or a mat of fine copper wires lying perpendicular to the flow direction and held between two screens. The regenerator is a poor conductor of heat so that even though there is a temperature gradient along it, there is a negligible flow of heat along the cylinder axis. The cylinder is insulated except for contact with the "hot" source (T_1) at one end and the "cold" reservoir (T_2) at the other. The operation of the engine is as follows.

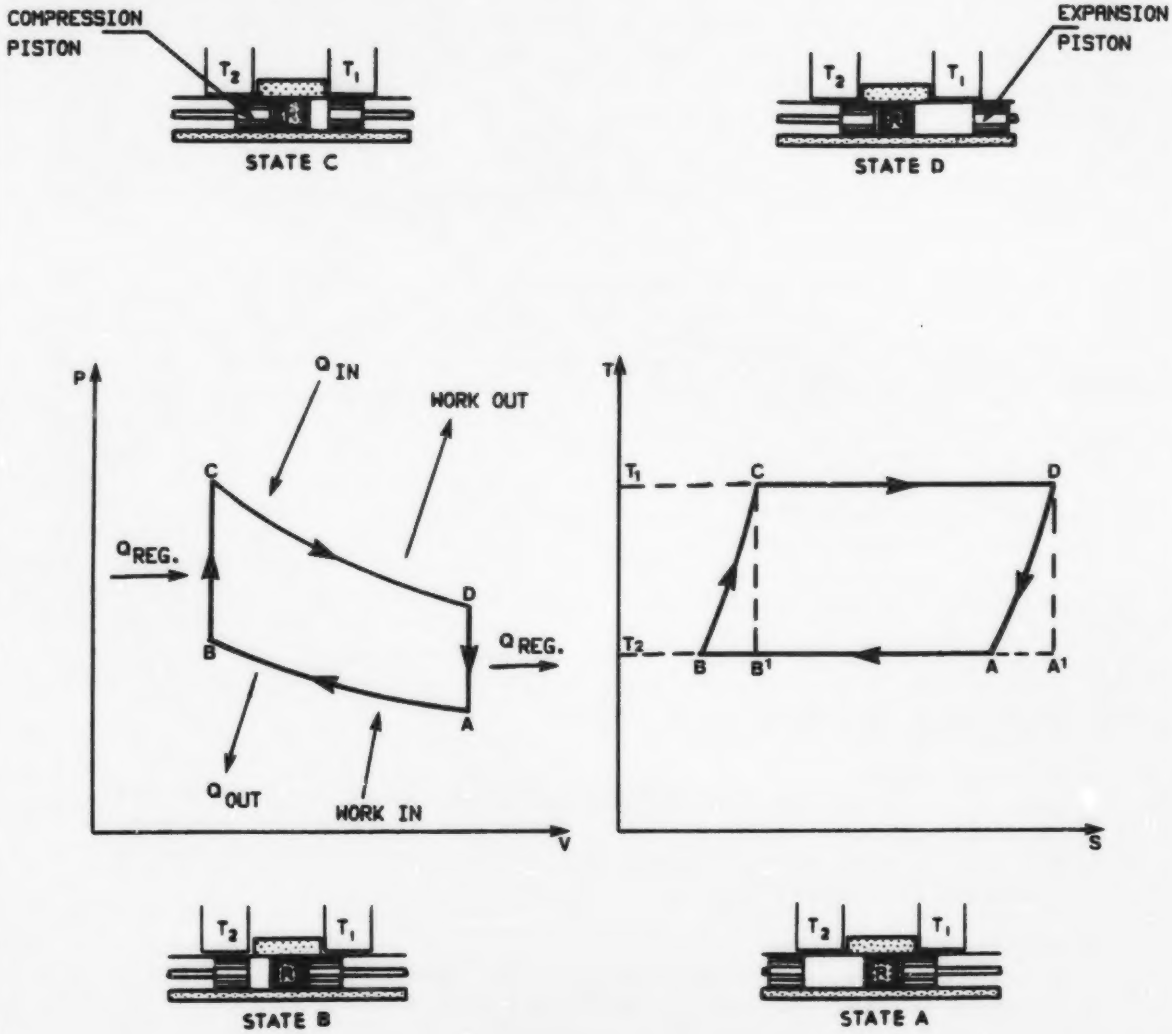
- Process A to B
  Heat Q_{out} is removed from the gas contained in the cylinder to the reservoir at T_2. In order to hold the gas temperature constant, the left piston moves inwards, work being done on the system as the pressure increases.

- Process B to C
  Both pistons move to the right at the same rate to maintain a constant volume. No heat is exchanged with either reservoir. As the gas passes through the regenerator it is heated to the temperature T_1. No work is done during this process since it is at constant volume.

- Process C to D
  Heat Q_{in} is transferred from the hot source at T_1. The right hand piston moves outwards, doing work as the pressure falls.

FIGURE 10.26

DIAGRAMMATIC REPRESENTATION OF THE STERLING CYCLE IN DUAL - PISTON ARRANGEMENT


COMPRESSION PISTON
T_2
T_1
STATE C
EXPANSION PISTON
T_2
T_1
STATE D
P
Q_{IN}
WORK OUT
C
$Q_{REG.}$
D
B
$Q_{REG.}$
A
Q_{OUT}
WORK IN
V
T
T_1
C
D
T_2
B
B^1
A
A^1
S
T_2
T_1
STATE B
T_2
T_1
STATE A

Q_{REG} - HEAT TRANSFERED TO/FROM REGENERATOR

▨ REGENERATOR

▨ CYLINDER INSULATION

[147]

- Process D to A
  Both pistons move to the left at the same rate, to keep the volume constant, and no heat transfer takes place with either reservoir. The gas is cooled as it passes through the regenerator and reaches temperature T_2. No work is done during this process since it is at constant volume.

This illustration of the Stirling cycle is pure theory. It must be remembered that the cycle diagram is an artifice for an abstraction in the same sense that an Otto cycle diagram is an abstraction in relation to the automotive engine.

The above definition and cycle description is sufficiently vague to cover any number of quite different engine designs and a great many have been proposed.

In theory the Stirling engine can achieve very high efficiency, i.e., equal to that of the Carnot cycle. It can be seen from Figure 10.26 that paths B-C and D-A are parallel, and the cycle can be "squared off" to yield the Carnot cycle shown as A^1-B^1-C-D. The inability of actual engines to achieve this limiting efficiency arises principally from the heat transfer process and pressure losses in the engine and its associated piping. One of the main difficulties is the design of an efficient regenerator of reasonable size which can operate at a sufficiently high temperature.

One end of the regenerator must be continuously subject to the heat source temperature T_1 and this temperature should be as high as possible in order to achieve the highest efficiency. It is limited, however, by metallurgical problems. The size of the regenerator should be small in order to maintain the dead space volume as small as possible. The dead space decreases the maximum pressure in the cycle and hence also decreases the power output of the engine.

There are two main types of Stirling cycle engine arrangements:

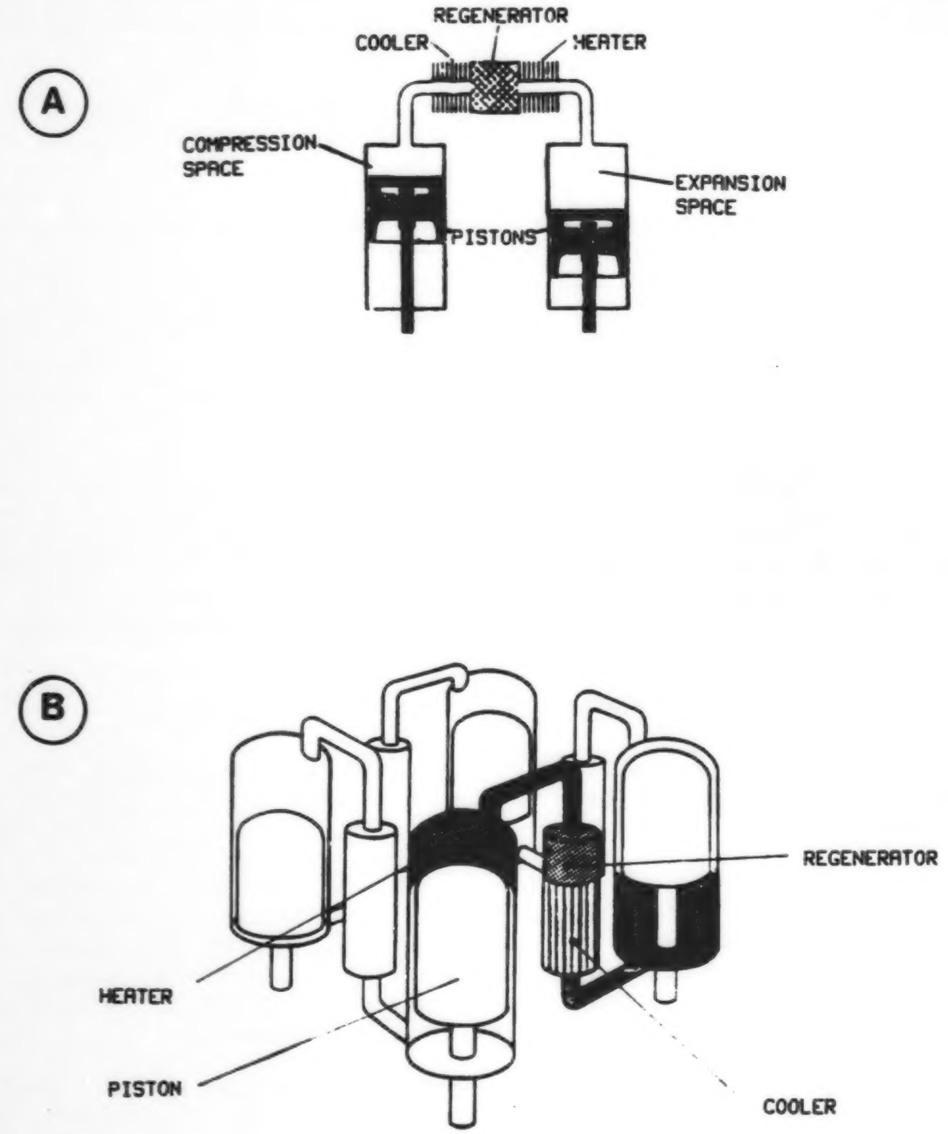
- Dual piston as shown in Figure 10.26 where separate compression and expansion pistons operate in the same or in adjacent cylinders.

- Piston displacer where both compression and expansion processes are brought about by the same piston. The displacer, as its name implies, simply shuttles the working fluid, at the appropriate points in the cycle, between the compression and expansion spaces. The piston and the displacer have to move out of phase to maintain such operation, therefore a complex linkage is necessary.

Within these two types there are many different versions of this engine. Typical constructions are shown in Figures 10.27 and 10.28 [148, 34]. Version A shown in Figure 10.27 presents a design with single acting pistons in separate cylinders, while version B is the multi-cylinder double acting "Rinia" configuration suitable for extremely compact high power systems. The four parallel cylinders can be arranged in a square with the pistons driving a swash plate. Each pair of adjacent cylinders forms a self-contained engine. The engine shown in Figure 10.28 uses the so called

FIGURE 10.27

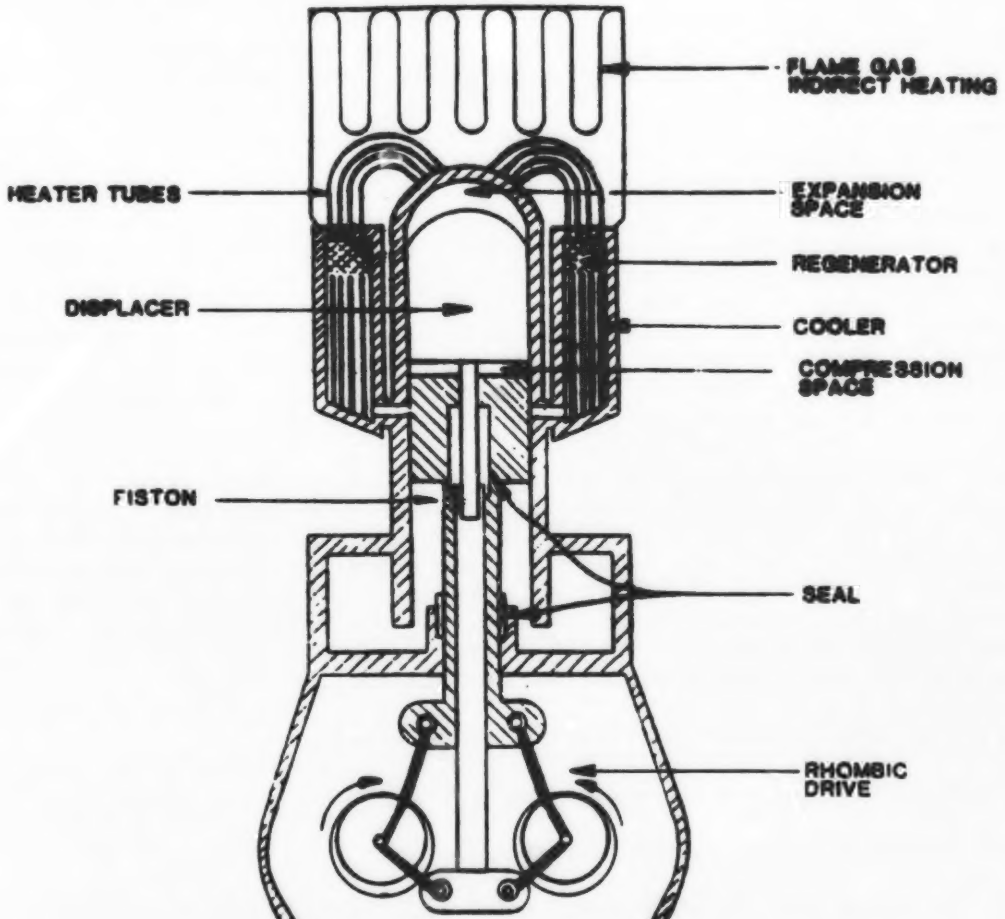
TWO VERSIONS OF STIRLING CYCLE ENGINE


A
REGENERATOR
COOLER
HEATER
COMPRESSION SPACE
EXPANSION SPACE
PISTONS
B
REGENERATOR
HEATER
PISTON
COOLER

[148]

FIGURE 10.28

PHILIPS STERLING CYCLE ENGINE
WITH RHOMBIC DRIVE


FLAME GAS INDIRECT HEATING
HEATER TUBES
EXPANSION SPACE
REGENERATOR
DISPLACER
COOLER
COMPRESSION SPACE
FISTON
SEAL
RHOMBIC DRIVE

[34]

Philips rhombic drive mechanism and displacer piston type arrangement. The drive comprises twin identical offset crank and connecting rod mechanisms rotating in opposite directions and connected by two synchronizing gear wheels. This engine is characterized by vibrationless and virtually silent work due to the possibility of the complete dynamic balance of all harmonics.

Many other versions of the Stirling engine have also been designed. Among them, a free piston engine well suited for water pumping and gas compression requirements and a version based on the Wankel principle.

Since the Stirling cycle machine operates on a closed cycle any inert gas can be used as a working medium. Hydrogen is often used for smaller engines and helium or air for bigger engines.

The location of an external heat source on the engine depends mainly on the engine size. Small engines have the combustion chamber directly over the cylinder heads. Heat is transferred through the short distance by means of numerous pipes or fins which increase heat transfer area. Direct heating is difficult to design for larger engines especially when coal firing is desired. That is why the combustion chamber is fitted separately but adjacent to the engine. Heat is supplied through the system of extended high pressure tubes, called a heat loop, which contain thermal fluid transporting heat. This arrangement reduces the engine efficiency but on the other hand allows the use of coal and other fuels as well as the operation of the multi-cylinder engine from one remotely located combustion chamber. In the case of coal firing, a mechanical stoker furnace or fluidized bed combustor can be used. Heat from a nuclear reactor can also be used as a heat source for this engine.

The present generation of Stirling engines have relatively poor efficiency and about two thirds of the supplied heat has to be rejected. In a Stirling cycle engine there is no possibility of rejecting the cooling load by an exhaust pipe due to the closed cycle, therefore the heat must be dissipated through the cooler. It means that the magnitude of cooling load of the cooler has to be about twice the load of the diesel engine of the same power. That is a problem for an automotive engine, but not for a marine application, where sea water can be used for cooling.

The major advantages of the Stirling engine are:

- ability to burn a variety of fuels
- theoretical high thermal efficiency
- low noise and vibration level mainly due to the lack of valves and high degree of balancing
- low exhaust emission level

- potential for engine long life and low lubrication oil consumption
- excellent part load performance, since the power of the engine can be regulated very simply by varying the mass of the working medium in the cycle which does not affect the efficiency of the cycle.

Despite these advantages, the Stirling cycle engine is not produced commercially because of the following disadvantages:

- high cost and complexity of its advanced high performance form
- high cycle temperature, necessary for obtaining high efficiency, requires special materials to withstand the operating conditions, seals are especially a problem
- very high capacity cooling system required.

It is expected that the application of Stirling cycle engines in the near future will be limited to those areas where no alternatives exist. The engine requires to be developed but has the potential to become one of the more serious competitors to the internal combustion engine.

Some of the present studies suggest it is now possible to build a Stirling cycle engine of 1,500 kW with an efficiency, size and weight similar to those of diesel engines of comparable size. Cost is expected to range from 25% to 50% greater than for the diesel engine. Some scientists claim that an engine thermal efficiency of 60% can be achieved.

The application of these engines to marine propulsion plants depends on their degree of development in the future. The multi-cylinder double acting engine configuration would be the most suitable for ship use and engines in the power range from 500 to 10,000 kW should be possible. The ships in which the Stirling engine could be used would be coastal ships, small tankers, ferries, tugs and other special purpose vessels [34].

10.10 Fuel Cells

A fuel cell is an electro-chemical device in which the chemical energy of a conventional fuel is converted directly into low voltage, direct current electrical energy. This cell eliminates the conventional fuel to heat to mechanical to electrical energy process by the use of an electro-chemical reaction similar to a storage battery process.

In a conventional battery the chemical energy is stored within the cell. Fuel cells are unlike batteries in that the reactants are supplied from outside the cell and the cell itself does not undergo an irreversible chemical change, i.e. the fuel cell components, electrodes and electrolytes, are not consumed in the energy conversion process.

The principle underlying any combustion reaction, whether in a fuel cell or a heat engine, is the same. The combustion reaction involves a transfer of electrons from the fuel molecules, thus oxidizing them, to the oxidizer molecules, thus reducing them. In the heat engine, fuel and oxidizer molecules are intimately mixed so that electrons pass directly from fuel molecules to oxidizer molecules. The fuel cell is a device that keeps the fuel molecules from mixing with the oxidizer molecules, permitting, however, the transfer of electrons by a metallic path that may contain an electric load.

During the past several years many different fuel cell systems have been built. The most useful and interesting systems are those which operate on more reactive fuels such as hydrogen, alcohol or hydrazine and on oxidizers such as oxygen or hydrogen peroxide. The most advanced type is the Hydrogen-Oxygen Cell which has been widely used for space applications. The Hydrogen-Oxygen Fuel Cell is shown in Figure 10.29. The fuel cell consists of a fuel electrode (anode), oxidant electrode (cathode) and an electrolyte. In this typical application hydrogen is fed through the porous electrode where it reacts in the presence of a catalyst with the hydroxyl ions of the electrolyte, potassium hydroxide. An electron interchange occurs at the electodes which results in an external load current. At the anode:

$$H_2 + 2OH^- \qquad 2H_2O + 2e^-$$

At the cathode the oxygen acquires electrons, which have been released at the anode and have been transferred due to the external current and reacts with water:

$$2e^- + O_2 + H_2O \qquad 2OH^-$$

A water molecule is the net product of the overall reaction:

$$2H_2 + O_2 \qquad 2H_2O$$

The electrical output from an individual cell is typically a direct current of 100 to 200 mA/cm^2 at a voltage of about 1 volt.

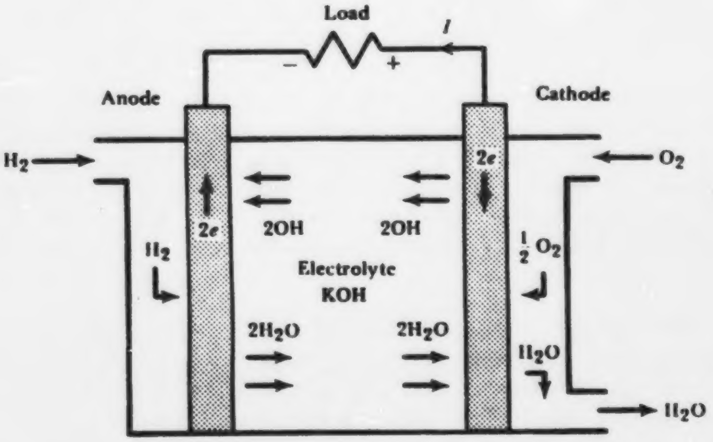
Many different types of fuel cells are under development. Some are at a rather advanced stage, others are still little more than laboratory curiosities. A substantial development effort has been directed toward producing low cost fuel cell that uses more readily available fuels, such as hydrocarbons or coal. The development of this type of fuel cell has been considerably more difficult than that of Hydrogen-Oxygen Cells. Common hydrocarbon fuels such as methane and petroleum derivatives are much less reactive than hydrogen and are therefore more difficult to oxidize. In addition, many hydrocarbon fuels, when oxidized, result in undesirable by-products which interfere with the operation of fuel cells.

There are several methods of classifying fuel cells: by type of fuel, oxidant, operating temperature, type of electrolyte, etc. For example:

- aqueous acid electolyte cells
- aqueous alkaline electrolyte cells
- molten salt electrolyte cells
- solid electrolyte cells

FIGURE 10.29

HYDROGEN - OXYGEN FUEL CELL


Load
I
−
+
Anode
Cathode
H_2
O_2
2e
2e
2OH
2OH
H_2
$\frac{1}{2} O_2$
Electrolyte
KOH
$2H_2O$
$2H_2O$
H_2O
H_2O

[149]

More general classification, used in literature [149], distinguishes between direct, indirect and regenerative fuel cells. In direct fuel cells the hydrogen or hydrogen containing fuel is fed directly to the anode. The indirect fuel cell utilizes the reforming process outside the cell, or internally at the anode, to convert the fuel to hydrogen which is then fed to the anode. The regenerative fuel cell is one in which the fuel cell product is reconverted into its reactants and recycled.

The Hydrogen-Oxygen Fuel Cell is the one of the direct types of cells. This cell is the most highly developed, since it employs simple chemical reactions and produces only pure water with minimal heat generation. The cell requires only a small amount of noble metal electro-catalyst which, in other types of fuel cells, is the most expensive component. The electrodes are the heart of the fuel cell. They must conduct electricity well and must be resistant to the highly corrosive environment. Various materials may be used to make electrodes, but the most common is platinum or similar precious metals, the use of which is restricted for economic reasons.

Various hydrocarbons, besides hydrogen, are potential fuels for fuel cells as well as the "compromise fuels", namely methanol, ammonia and hydrazine. "Compromise fuels" are generally easier to use and their reactivity is better than that of hydrocarbons. In view of future prospects, the use of these fuels in fuel cells is under intensive development.

Fuel cells developed for land utilities could be easily adapted to a shipboard installation. Some of the equipment necessary for a land operation would not be required for ship applications. Fuel cells could be suitable to power submersibles where low volume and high power density are required. It is also expected that fuel cells may offer cost advantages over conventional power systems in small vessels like coastal ships, ferries, tugs, etc.

In the case of larger ships utilizing fuel cells to drive an electric propulsion motor, the dimensions of the motor will be very large. The use of a superconducting electric machine would make the propulsion motor more manageable in weight and size and, additionally, will also improve plant efficiency by approximately 2.2%.

A fuel cell ship propulsion plant could be designed in the future as shown in Figure 10.30. This design has been proposed by Baham Corp. and is based on a flow diagram for a phosphoric fuel cell system which has been developed for a demonstration fuel cell plant in New York city by the Electric Power Research Institute and a group of private companies [34].

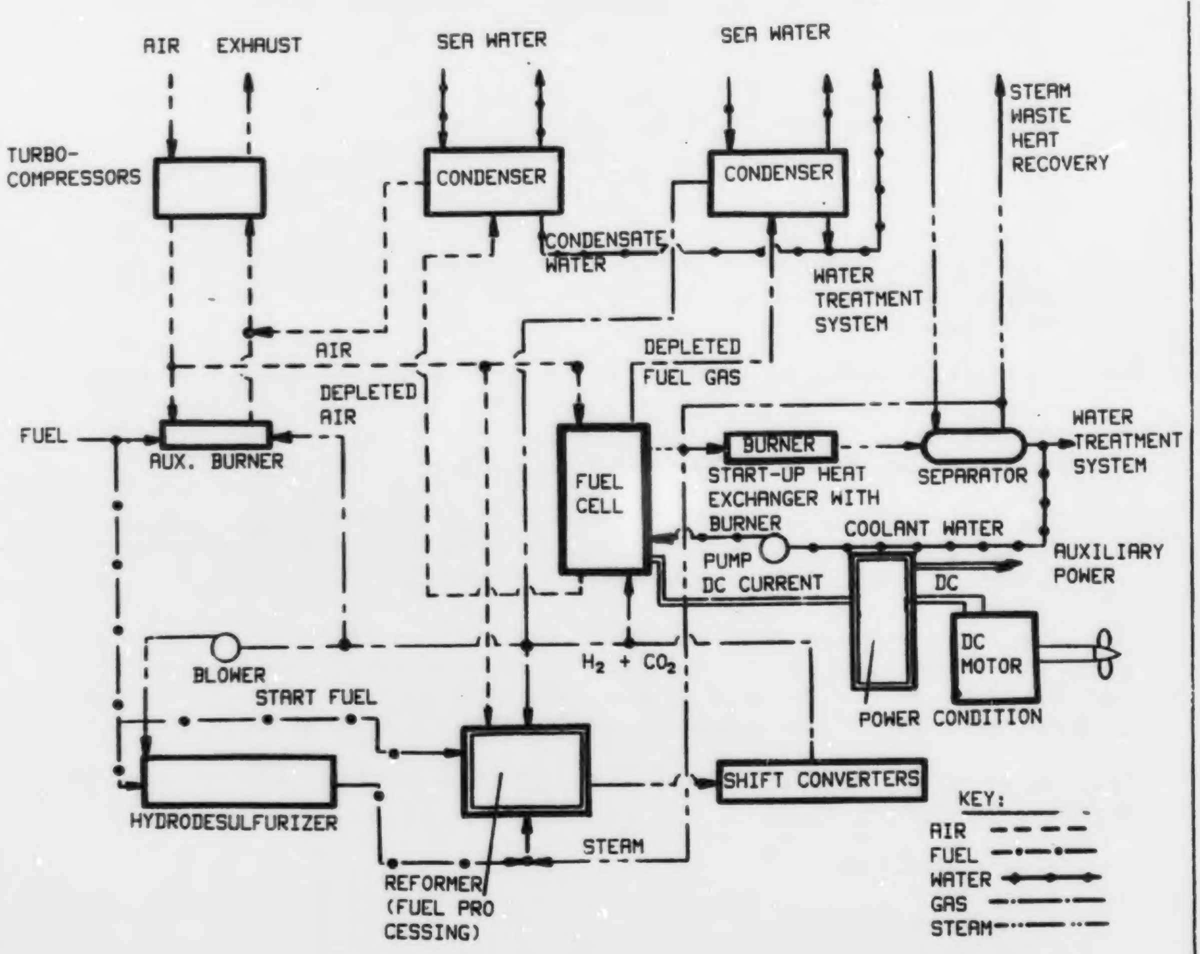
The functions of the main system components are as follows:

- The turbo compressor distributes the air to the cathodes to provide oxygen for the fuel cell reaction, to the fuel reformer and to an auxiliary burner which is used to add energy at low loads to the gas stream powering the turbo compressor.

- The condensers are used to condense water contained in the depleted air and fuel gas which leave the fuel cell. The condensed water is sent to the water treatment system to provide

FIGURE 10.30

MARINE FUEL CELL PLANT SCHEMATIC


AIR
EXHAUST
SEA WATER
SEA WATER
STEAM WASTE HEAT RECOVERY
TURBO-COMPRESSORS
CONDENSER
CONDENSER
CONDENSATE WATER
WATER TREATMENT SYSTEM
AIR
DEPLETED FUEL GAS
DEPLETED AIR
FUEL
AUX. BURNER
FUEL CELL
BURNER
START-UP HEAT EXCHANGER WITH BURNER
SEPARATOR
WATER TREATMENT SYSTEM
COOLANT WATER
PUMP
DC CURRENT
AUXILIARY POWER
DC
DC MOTOR
BLOWER
$H_2 + CO_2$
POWER CONDITION
START FUEL
SHIFT CONVERTERS
HYDRODESULFURIZER
STEAM
REFORMER (FUEL PROCESSING)
KEY:
AIR
FUEL
WATER
GAS
STEAM

[34]

make-up water for fuel cell. The dry air is used to help power the turbo compressor and the dry exhaust gas enters the fuel reformer burner where the remaining dilute hydrogen is burned to provide heat for the reforming process.

- The reformer converts the fuel to a mixture of hydrogen, carbon dioxide, carbon monoxide and water vapour. Prior to the reformer, fuel is mixed with steam from the power section waste heat system.

- The shift converters convert the carbon monoxide and water from the reformer to hydrogen and carbon dioxide. Less than one percent of carbon dioxide is present in the mixture after the converters. This mixture goes then to the anode cavities in the fuel cell.

- The hydro-desulphurizer changes sulphur compounds to hydrogen sulphide and absorbs it in a zinc oxide bed.

- The fuel supply by means of a fuel pump is mainly directed to the hydro-desulpherizer prior to which is mixed with hydrogen rich gas from the fuel processor. During startup the part of the fuel is sent to the reformer to burn, and at reduced power, some fuel is sent to the auxiliary burner in order to power the turbo-compressor.

- Fuel cell cooling is achieved by means of the water pumped in the cooler. Water is partially converted to steam which is then removed by a separator. A portion of the steam is sent to the reformer and the remainder goes to a waste heat recovery unit.

- The DC power output coming directly from the fuel cells may be converted by an inverter to AC power depending on the electrical design of the propulsion and auxiliary systems.

The problem of voltage regulation and the ability of the fuel cell power plant to follow variations of load is not yet well developed and needs further research. It is particularly important in ship application because of more rapid and extreme power variations than in utility power stations.

Fuel cells offer some advantages over other electrical energy conversion systems, namely:

- very efficient
- quiet
- simple and compact
- reliable
- non-polluting

The high conversion efficiency is the main advantage. Fuel cells are not subject to the Carnot limitation of efficiency as other conventional systems of energy conversion. Since the process in the fuel cell is carried out isothermally, at least theoretically, its efficiency is determined only by the thermo-dynamics of the cell chemical reactions. The energy generation in the fuel cell is a one-step process, hence it does not create any heat and mechanical losses as a heat engine. The fuel cell has no moving parts, except for possible auxiliary pumps, and although some heat is generated in a fuel cell, it has no important affect in the energy conversion process.

The efficiency of a Hydrogen-Oxygen Cell calculated for theoretical conditions, i.e., for a reversible, isothermal, constant pressure process and for a temperature of 25°C, is 83%. Obviously, the actual efficiency is lower, but it has been proved that hydrocarbon fuel cells operate at much the same efficiency for small units in the 100 kW range to large multi-megawatt units. The efficiency of a practical fuel cell system is approximately 55% if pure hydrogen is used as a fuel, which can be increased to nearly 60% efficient if oxygen is used instead of air as an oxidizer.

In types of fuel cells which give off larger amounts of heat during operation, the overall efficiency may be improved by utilizing the waste heat to process fuel or for other purposes.

The most complicated hydrocarbon fuel cell is still more simple than a heat engine generator. Controls are necessary but these are not complicated and can be automated. Fuel cells can be connected into large cell stacks without loss of efficiency, as their efficiency remains almost constant over a wide range of power. Since build-up of a "power pack" is made by multiplication of identical cells, cost savings can be expected due to the use of mass production.

Unfortunately, despite obvious advantages, fuel cells are still not widely used as practical energy sources. This is because the cells are expensive, principally due to the degradation of electrode materials, and have a short life cycle. The other very important disadvantage of fuel cells is that they cannot use coal directly as a fuel. The use of coal derived fuels in the future may increase the price advantage of fuel cells. These are the major problems to be overcome if fuel cells are to be commercial viable.

Present research and development of fuel cells are directed towards their use in space and utility applications. The most significant research has been conducted by such companies as The Pratt and Whitney Aircraft Division of United Technologies Corporation, Esso-Alsthom and Westinghouse. They, and a number of other research centres, have developed the existing technology for fuel cell power systems. Further development will undoubtedly lead to improvement in fuel cell technology and increased utilization.

The economics of a fuel cell based marine propulsion systems are not firmly established and costs are difficult to obtain. Some idea can be obtained by review of other commercial fuel cell system costs. The most expensive are the aerospace fuel cells which cost in the range of $10,000 to $400,000 per kW in 1978 U.S. $. For stationary hydrocarbon fuel cell power systems the expected costs vary from $300 to $1,000 per kW. The estimated costs of some fuel cell systems and produced energy presented in Baham Corp. study are shown in Table 10.6 [34].

Roughly estimated these total energy costs are over twice the costs of the energy for conventional power plants. It is expected, however, that improvements in efficiency and mass produced fuel cell components will make the energy cost competitive in the future.

TABLE 10.6

FUEL CELL COSTS AND WEIGHT
(1979 U.S.$)

FUEL CELL ELECTROLYTE	CAPITAL COST $/kW	TOTAL ENERGY COST ¢/kWh	PLANT WEIGHT kg/kW
Phosphoric Acid	440 - 560	5.0 - 6.5	-
Potassium Hydroxide	565 - 875	5.5 - 6.0	4.0
Molten Carbonate	660 - 780	4.5 - 9.0	-
Stabilized Zirconia	525 - 1180	4.0 - 8.0	-

[34]

10.11 Magnetohydrodynamic Power Generation

The basic concept of the magnetohydrodynamic (MHD) converter is exactly the same as the principle of all rotating electrical generators in which the rotor turns conductors in a magnetic field to produce electricity. In the case of MHD generator the conductor is replaced by a high speed ionized gas called a plasma. The electrical properties of a plasma are determined by the density of free electrons in the gas. The ionization of the gas, and consequently its electrical conductivity, can be achieved in several ways:

- high gas temperatures
- seeding of a gas with readily ionized atoms such as caesium
- nuclear radiation of the gas and others.

When the ionized gas flows at right angles to the magnetic field, a force is generated at right angles to the direction of the gas flow and the magnetic field. This force drives the electrons and ions to the pickup electrodes, and gives rise to a current when the electrodes are connected to an external load.

In the MHD cycle the gas is first compressed, heat is added at constant pressure, and the gas is then accelerated in a nozzle before passing through the magnetic field. Considering the overall cycle such as the Rankine or Brayton cycle it can be seen that the MHD generator replaces the turbine and generator. For this reason the MHD generator is a simpler device than a conventional turbo-generator power plant. The lack of highly stressed moving parts in the generator makes mechanical tolerances relatively broad and permits higher gas temperatures to be used than in conventional turbomachinery.

Three different types of MHD generators are identified. The simplest is the open cycle MHD system. In it the generator fluid is the high temperature combustion gas obtained from burning a fossil fuel or coal. The gas is "seeded" usually with potassium salts to give high level of ionisation and hence electrical conductivity.

The other plasma system is a closed cycle MHD in which a very pure inert gas is raised to high temperature in a heat exchanger and as the previous system is "seeded" with a pure alkali metal. A nuclear reactor or an externally fired heater can be used as the heat exchanger.

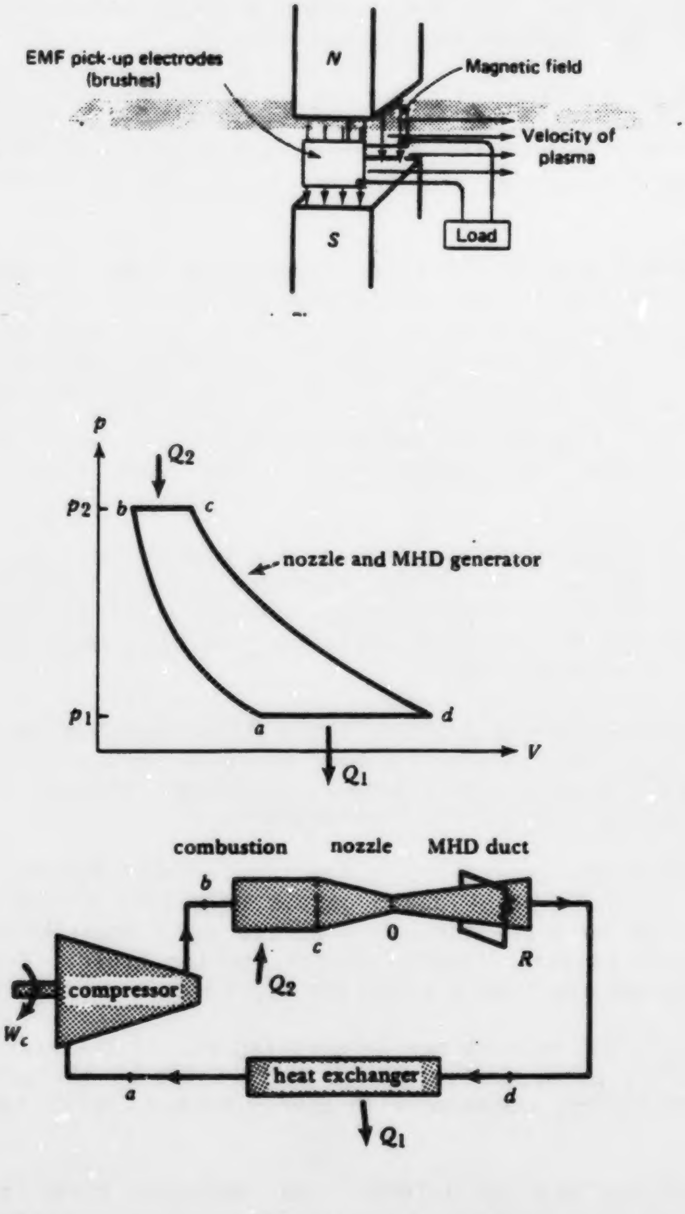
The third type of MHD generators is the liquid metal system. In this system a mixture of a liquid metal and a gas is raised to a high temperature and expanded to high velocity through a nozzle as a foamlike substance. This foamlike mixture is used directly as the working fluid. After exciting the generator, the gas and liquid metal are separated and recycled.

The closed cycle MHD systems are interesting as, if the working fluid can be kept sufficiently pure, equivalent conductivities of the working fluids can be obtained at 1650°C, compared with approximately 2500°C for the open cycle systems.

MHD power systems are of interest for advanced power plants primarily because of their high performance potentials. Their performance is directly related to their maximum temperatures. Since open cycle systems operate with the highest temperatures, they have the highest performance potential

FIGURE 10.31

MAGNETOHYDRODYNAMIC GENERATOR PRINCIPLE AND ITS IDEAL CYCLE


EMF pick-up electrodes (brushes)
N
S
Magnetic field
Velocity of plasma
Load
p
Q_2
p_2
b
c
nozzle and MHD generator
p_1
a
d
V
Q_1
combustion
nozzle
MHD duct
b
c
0
R
Q_2
compressor
W_c
heat exchanger
a
d
Q_1

[150]

and the promise of high efficiency. However, this device is a form of heat engine and is subject to the efficiency limitations of the Carnot cycle. Figure 10.31 shows the principle of MHD generator and the basic theoretical closed cycle.

A great deal of research into MHD generators was carried out in U.S.A., U.K., F.R. of Germany, Japan and some other countries during the last 15 years. The United States has considered MHD for ship propulsion since 1960. The compactness and the absence of intermediate rotating machinery were considered to be important from the viewpoint of reducing noise, especially in submarines. The MHD generator is simple but when it is combined with steam or gas turbine cycle it can become complicated. All of the existing components of a conventional steam turbine plant would still exist. The MHD generator with all its auxiliaries and maintenance and operation problems would be additional to those of a conventional power plant [34].

Because of the complex lay out, MHD systems are estimated to be expensive. In addition, since MHD systems produce direct current power, they require costly inverters to convert this power for alternating current transmission.

Despite bright prospects, MHD systems are still one of the least developed concepts. There are many unknowns concerning components and plant design as well as costs. It is claimed that for large utility power stations the cost of electricity could be competitive in the near future. The application of MHD generators in the marine field looks less likely. It is expected that the technology would be available for marine plants not sooner than the year 2,000.

10.12 Thermionic Converter

A thermionic converter is a high temperature device that converts thermal energy to electrical energy by utilizing the phonemonon of thermionic emission. In the simplest form the converter consists of two electrode surfaces, one of which (called the emitter) is heated, whereas the other (called the collector) is cooled. Electrons are given sufficient thermal energy in the emitter to allow some of them to escape and travel to the collector across the short distance of vacuum or plasma. The electrons then return to the emitter through an external load circuit, thus producing electrical power.

The thermionic converter can be regarded as a heat engine that uses an electron gas as a working substance. Thus the upper limit of its efficiency is that of a Carnot engine operating between the temperature limits of the emitter and collector. Therefore a high temperature emitter is required to yield high efficiency. The high emitter temperature is also advantageous because electron emission is an exponential function of temperature, thus higher temperature gives higher power densities.

The efficient generation of electrical power is inhibited by source problems, however. The main drawback is the influence of space charge effect on electron flow. The emission of electrons from the hotter electrode is inhibited by space charge resulting from the accumulation of the electrons in its vicinity. To counteract this, the gap between electrodes should be as small as 0.02 mm, otherwise the retarding potential of the space charge becomes too large for satisfactory operation of the converter.

Filling the interelectrode space with a positively charged gas also facilitates the emission of electrons and allows an increase of distance between the electrodes to 1.0 mm. Substantial losses due to the heat radiated and conducted from the emitter to the collector may be incured.

Clearly mechanical design, especially appropriate electrode spacing with large temperature differences between the electodes plays the most important role in the ultimate performance of a thermionic converter.

Efficiencies of these converters are rather low. With an emitter temperature of 1700°C, efficiency of 10% has been achieved and at 2000°C, about 18%. The ranges of output voltage have been noted as 0.2 - 0.4 volt and 0.5 - 1.0 respectively.

Due to the possibility of producing electricity directly without any moving components the initial applications of a thermionic converters are to space craft.

These converters have also been proposed as topping units for stationary and marine power plants as part of a combined-cycle generation systems. Hot combustion gases would supply heat to the emitter at about 1,700°C and the heat tabeu from the collector at around 550^{C} would produce superheated steam to operate a conventional turbo generator.

By these means improvement in overall efficiency for conversion of heat into electricity of 3 to 105 may be achieved. The extra cost of the additional equipment is expected to be repaid in 2 to 5 years depending on the size of installation.

The current research and development work in this area and in the field of materials which can withstand the high temperatures required, give prospects for the thermionic cycle to be applied to marine applications not sooner than in 10-15 years.

10.13 Summary

The wide range of options offers prospects for development of future propulsions systems and ship machinery. Some of the options discussed have the immediate prospect of adoption by the marine industry, other proposed ideas only have a remote potential of being translated into practical design.

Some types of coal fired propulsion systems have already been introduced on vessels recently built. Their economic success depends mostly on the price relationship between coal and oil. At the present time coal is not competitive with oil for ships' bunkers, but when this relationship changes, the proven technology will be advantageous for those shipowners involved in these experimental coal fired ships.

The only systems available at present, for the commrrcial conversion of coal to energy, are coal fired boilers providing steam for turbines. The alternative to this sytem is the use of coal as a fuel for closed cycle gas turbine systems, which will provide a far more economic use of the fuel and allow higher system efficiencies to be gained. Similarly, the Stirling engine using coal as a fuel also has potential but it would seem that the gas turbine is in a more advanced stage of development than high powered Stirling engines.

The development of fluidized bed boilers or combustors is believed to be an important component in the general development of more efficient heat engines capable of operating with low grade fuels. It is possible that the successful demonstration of a fluidized bed boiler, suitable for shipboard operation will be the stimulus required for accelerating the work on advanced ship propulsion systems such as the Stirling engine, closed cycle gas turbines and other topping cycles. The development of these advanced heat engines depends very much upon the availability of suitable materials for high temperature service.

The use of nuclear plants, except for special applications, is not expected to be justifiable for general ship propulsion use. The special applications where nuclear vessels could be justified are the high powered ships which might be required for use in the arctic. Vessels powered by nuclear boilers are already in use in the U.S.S.R. for this purpose. There is considerable opposition in the western democracies to the use of nuclear powered ships in any capacity, based upon environmental considerations, but there are no technical reasons why these vessels should not be built. The technology for nuclear vessels is available and has been used extensively for war ships. The capital costs and operating costs of a nuclear powered ship are uncertain, as the price of uranium fuel is governed more by political considerations than by free market forces. It is expected however, that nuclear fuels will in the future be in short supply and the use of exotic fast breeder reactors will perhaps be necessary. These reactors have not been used at sea and are in a state of early development ashore.

The use of fuel cells and other methods of direct conversion of chemical to electrical energy at sea is considered to be remote in the immediate future although cannot be discounted for the long term.

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